WORKSHOP PRACTICE

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PLAIN BEARINGS

BY

A. J. AIERS

BALL AND ROLLER BEARINGS

BY

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PETROL ENGINE AND GAS ENGINE FITTING AND ASSEMBLY

BY

MAJOR A. GARRARD, Wh.Ex.

WORKSHOP PRACTICE

A PRACTICAL WORK FOR THE DRAUGHTSMAN, THE MECHANIC, THE PATTERN MAKER, AND THE FOUNDRYMAN

EDITED BY

E. A. ATKINS

M.I.MECH.E., M.I W.E

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WORKSHOP PRACTICE

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PREFACE

THIS and the subsequent volume have been compiled expressly for the benefit of those mechanics whose duty it is to attend to the fitting, erection, and testing of internal combustion engines, steam engines, and steam turbines. The present volume deals with petrol engine and gas engine fitting and assembly. Preliminary to considering actual fitting and erection operations it has been deemed advisable to offer to the reader some useful matter on the subject of bearings which, of course, are of primary importance in connection with all engine work.

In Section XXX he will find a very complete treatment of plain bearings in which every conceivable type is touched on. Bearings for marine purposes, automobile work, stationary engines, locomotives, and the numerous other purposes are all dealt with. The author also deals with oil-less bushes, such as are used in watches, magnetos, and other delicate precision instruments. Bearing construction, mounting and fitting are all clearly described, and the lubrication of each type, needless to say, receives the thorough treatment which is deserved. Various types of oil pumps and indicators are discussed, and the recognized systems of oil grooves explained.

Ball and roller bearings are the subject matter of Section XXXI. These bearings, which are used where loading and speeds are high and where the elimination of friction is of paramount importance, are very fully described and illustrated. The fundamental difference between plain bearings and ball or roller bearings, i.e. the substitution of rolling friction for sliding friction

forms the essence of the author's introductory remarks He thereafter proceeds to describe the principles construction, and mounting of the Hoffmann, Skefkc Hyatt, and Timken bearings, and to detail their variou applications.

Major Garrard, an authority on internal combustio engines, in the two final sections in this volume, present in a most interesting and lucid manner the fruits of hi experience and knowledge of the fitting and erectio of internal combustion engines—petrol and gas engines These sections are very well illustrated by clear sketche and diagrams

Valve grinding, valve and ignition timing, piston rin lapping, bearings and their alignment, working clear ances, joint making, and lubrication are all in tur discussed, and emphasis laid upon the chief things to b noticed and attended to by the mechanic during th course of erection The author also deals briefly wit the matter of engine foundations.

On the petrol engine section, which doubtless wi prove decidedly valuable, the four-stroke engine alon is considered. This is excusable, since little differenc exists between the fitting and assembly of the two types except that in the case of the two-stroke the operation are rendered vastly more simple due to the entir absence of complicated valve gear. The two-strok engine, it should be remembered, is used mainly fc light-weight motor-cycles and for small stationar purposes.

The reasons given above, we think, justify th omission of special reference to this type of engine i the work.

The testing of I.C. engines is dealt with in anothe volume.

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SECTION XXX

PLAIN BEARINGS: TYPES AND FITTING

INTRODUCTION

BEARINGS are the most important parts of any machine, as upon their behaviour depends the efficiency, and also the output, of the machine. They act as contacts between the main frame and all the moving parts, and transmit rotating, sliding, and revolving motions. Whatever machine is designed, whether its function is for turning or machining metals, weaving, baking, transport on land, sea, and air, or road-making, bearings of one kind or another are necessary There are many bearings on each machine, and if one breaks down under service, the whole machine must stop. Bearings are also the parts of the machine which bear the friction, and a designer aims to cut this down to a minimum undue loss of strength. All machines or engines transfer energy from one place to another, and if the bearings are correctly designed the loss of power which is wasted by friction is not excessive. Everyone knows what bearings are, but the practical side of their uses and fitting is the least understood of all mechanical subjects. Again, most people know that a bearing requires a lubricant, but that is as far as general knowledge goes, therefore the author wishes to explain and illustrate the many varieties of bearings, and also their methods of lubrication and fitting.

FRICTION

Friction tends to stop the parts which we require to move, and the force extended in moving them

generates heat. This friction is very useful at times; fo instance, it is used to drive transport, such as the wheel of railway engines and motor-cars, all of which depen upon the friction between wheels and the rail or road

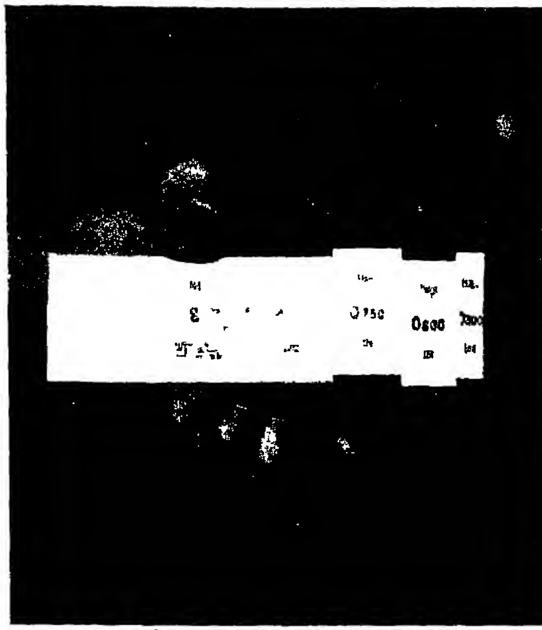


FIG 1. EXPERIMENT TO DEMONSTRATE HOW BODIES WITH PERFECTLY CLEAN AND SMOOTH SURFACE STICK TOGETHER BY FRICTION

Again, there are clutches and also belts for tran mitting-drives, but where bearings are concerned v want to cut down the friction. Solids tend to stic together if the contracting surfaces are dead smoo and free from any irregularities, owing to the fact th atmospheric pressure (14 lb. to the sq. in.) plays r

rôle between the two bodies, and thus the friction becomes very great when the bodies slide relatively to one another, and this is demonstrated in Fig. 1, where solid pieces of steel are supported only by friction, due to their surfaces being flat and clean. With bearings whose surfaces also have to be quite smooth, the

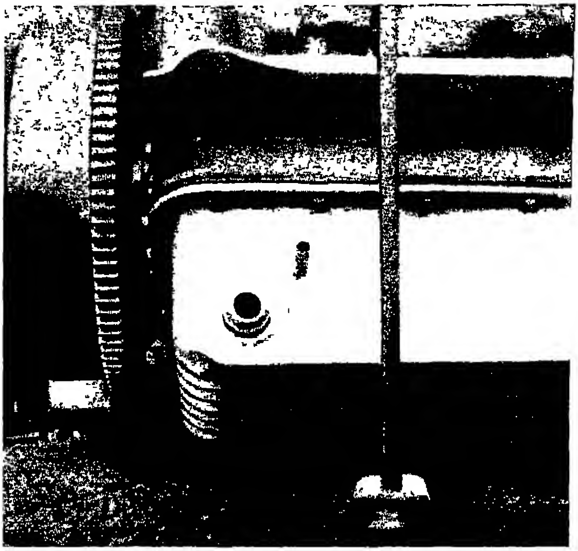


FIG. 2. COOLING FINS ON THE OIL SUMP OF A PETROL ENGINE

same thing holds good, and this is the reason for a lubricant, which acts as a layer between the surfaces and stops them from seizing. There is still a certain amount of friction and, therefore, heat generated, and, if excessive, the heat must be carried away by the lubricant and then the lubricant cooled. Fig. 2 illustrates the cooling fins on an oil sump of a petrol engine.

Not only must a bearing run cool, but it must not wear rapidly.

JOURNAL BEARING

Of the many types of bearings, the most well known and used is the journal bearing, Fig. 3, in which the load acts at right angles to the axis of the spindle.

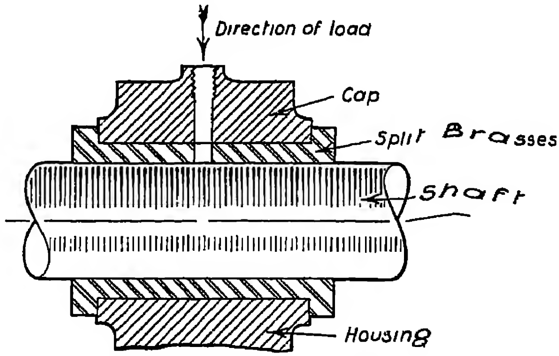


FIG. 3. SECTIONAL DRAWING OF JOURNAL BEARING

THRUST BEARINGS

In thrust bearings the load acts parallel to the axis, as illustrated in Fig. 4, where it is used in conjunction with a journal bearing. The two steel washers take care of the end thrust. If a spindle requires to carry a load and also take a thrust, a conical bearing could be used which would take care of both loads, but the friction is excessive, and usually separate journals and thrust bearings are fitted. When the shaft or spindle is in the vertical position, as in Fig. 6, the bearing is given the name of Footstep bearing. These are fitted with single or multiple thrust washers, according to the load to be carried. There are also many types of bearings

which only take a sliding motion, such as the bearing on the ram of a press (Fig. 7), the bores of cylinders and various sliding ways. Floating bushes are fitted where the housing and shaft are of suitable material. In these bushes, the fits of both the outside and the

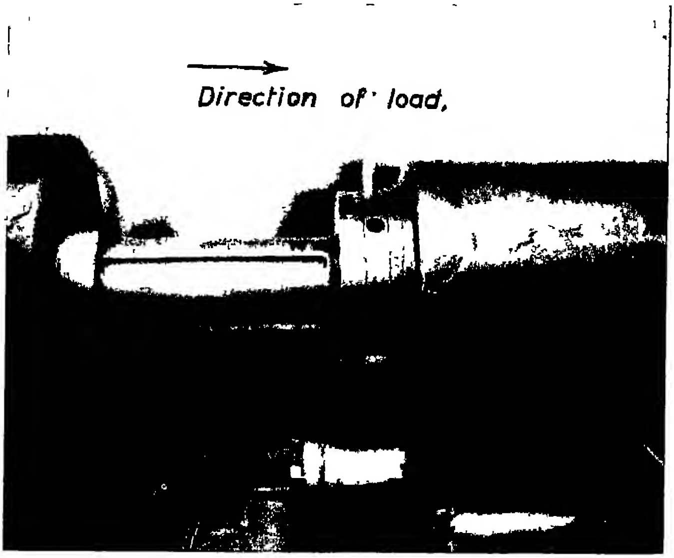


FIG 4. A TYPICAL THRUST BEARING

bore allow running clearance, but end pieces must be fitted to locate the bush.

On most machines it is necessary to transmit motion to slides, and this is usually accomplished by a screw and nut mechanism. In this case the nut is an important bearing, and one which is often overlooked. The usual material is bronze, but some are made by casting a white metal nut, using the screw as a mandrel.

SOLID AND ADJUSTABLE BEARINGS

All of the bearings mentioned are again divided int solid and adjustable types, though it is now generall

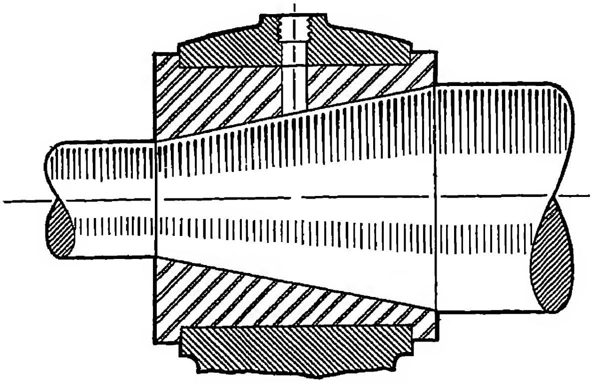


FIG. 5. CONICAL BEARING

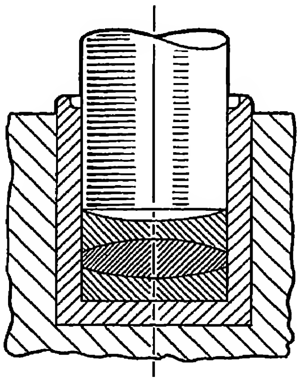


FIG. 6. SECTION OF FOOTSTEP BEARING

recognized that solid bearings are obsolete, except i special cases. A solid type, such as Fig 8, is sati factory as long as slight play does not affect the u

of the machine, but when wear takes place, a new bearing is necessary With adjustable bearings (Figs. 9 and 10), if play develops, it can soon be taken up by removing metal from the faces of the split bush. Special bearings are also made in which there are four pieces,

FIG 7 BEARING ON RAM OF PRESS

namely, top and bottom and side pieces, which can be adjusted by wedges

SPLIT BEARINGS

Another type of adjustable bearing is the tapered bush, and also the split bush. A tapered bush is illustrated in Figs. 11 and 12, and the adjustment is made by moving the bush endways, by means of screwed collars. Being split, the housing compresses the bush as it is forced farther in. Fig. 12 will show the reader how this adjustment is performed.

FIG. 8. SOLID TYPE BEARING

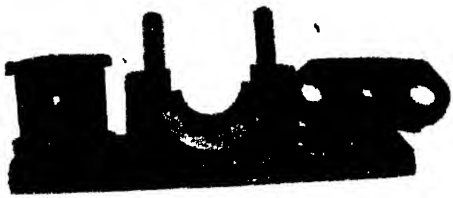


FIG. 9. SPLIT-BUSH TYPE BEARING

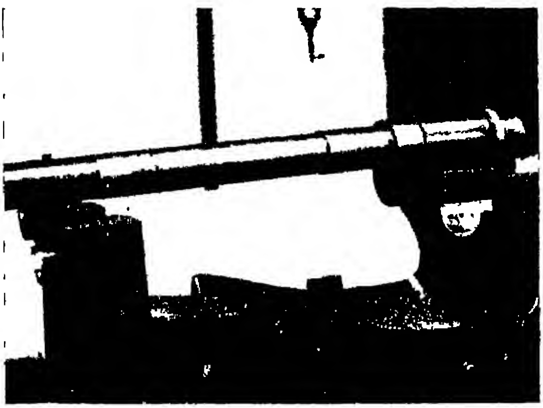


FIG. 10 BEARINGS AND SPINDLE OF LATHE HEADSTOCK

LATHE BEARINGS

With lathes and other machine tools, the main bearings must be a good fit, and must also be capable of being kept in this condition, therefore adjustment must be available as, otherwise, the play in the spindle

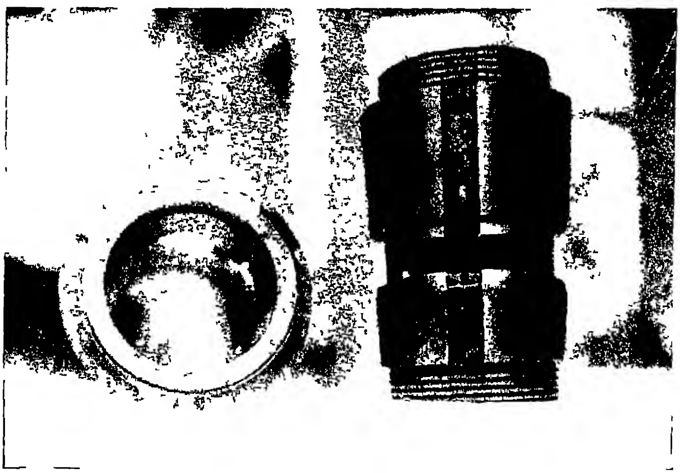


FIG 11 ADJUSTABLE TAPER BEARING

would not allow accurate work to be produced. As will be seen from Fig. 13, the main bearings are split so that play can be taken up. The bottom half of the brass rests in the headstock and is, of course, pegged to prevent rotation, while the top half is held down by a cap through which the lubricant is supplied Another headstock bearing is shown in Figs. 14 and 15, the bearing material in this case being white metal, and the thrust is taken on a thrust race fitted at the rear bearing. There are also the ways and beds of the machine, which are bearings where only sliding action takes place, but, nevertheless, they require oil grooves

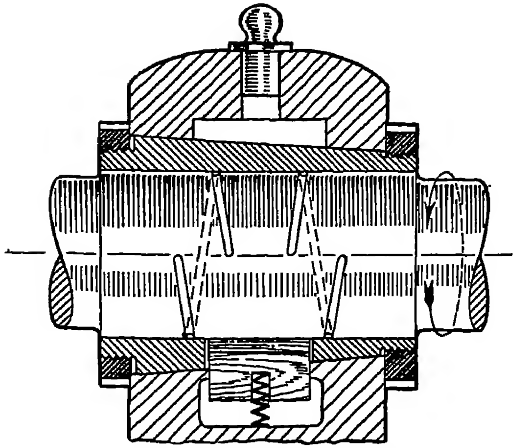


FIG. 12. ADJUSTMENT ON TAPERED BUSH BEARING

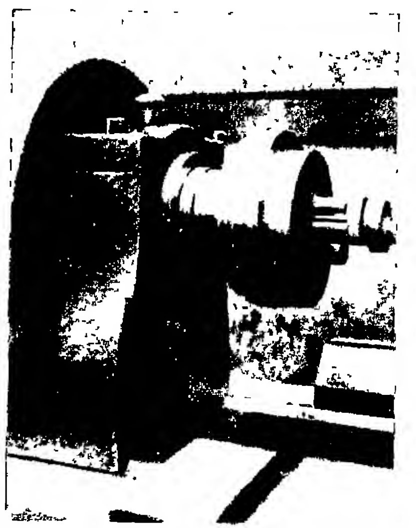


FIG. 13. BEARINGS OF LATHE HEADSTOCK

to distribute the oil, and also need provision to keep out turnings which would spoil the surface.

Some of these slides are extensive, as here the object is not to keep down friction so much as freedom from wear, and also stiffness to withstand the heavy cuts.

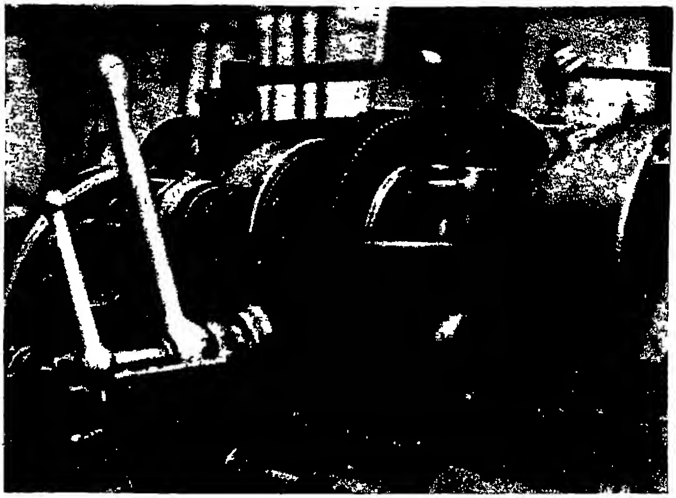


FIG 14. HEADSTOCK BEARING

Fig 17 shows one good method of fitting small loose pieces in front of all these surfaces, which hold a felt washer so that oil is kept in, and, a more important point, dirt and foreign matter kept out

SHAFTING BEARINGS (JOURNALS)

Although line shafting bearings now consist of either the ball or roller type, there is hardly a factory in which some of the shafting and most of the countershafts are not of the plain type. With shafting it is important that the various bearings are in line, and, to

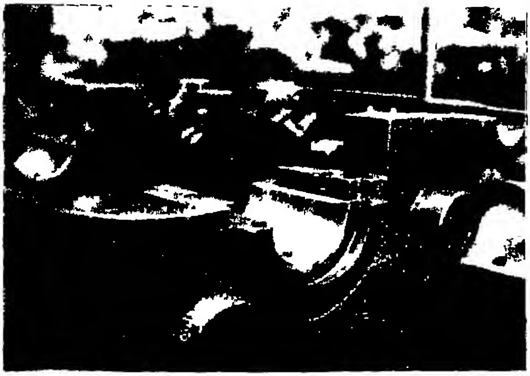


Fig. 15. Headstock Bearing Dismantled

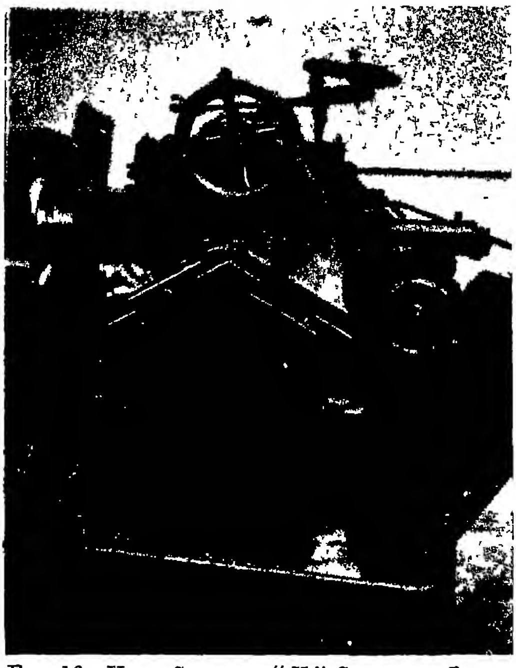


Fig. 16. View Showing "V" Slides of Lathe

facilitate this, they are adjustable in the hangars, either by screwed pins or other kinds of spherical mounting, all of which will be illustrated.

The methods of lubrication are usually by means of ring oilers, in which a slot is cut in the top of the bearing, and a loose ring slipped into this slot. As the shaft

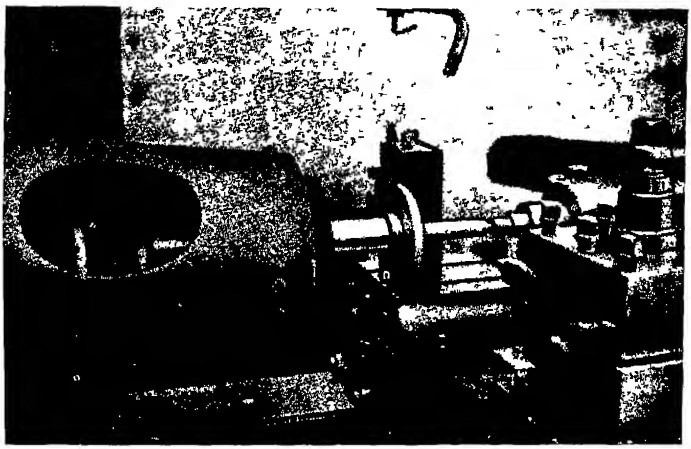


FIG. 17. SLIDE WITH PROTECTIVE SHIELD AND FELTWASHERS

revolves it turns the ring, the lower half of which dips in an oil bath and carries the oil up to the surface of the bearing. By viewing Fig. 8, and also the bearings in Figs. 18 and 19, the reader will thoroughly grasp this method.

In another type, instead of a loose ring, a collar is joined on the shaft which lifts the oil in a similar manner. This, of course, is more certain than a ring, and will operate at any speed Ring oiling is not satisfactory at very high speeds as slip occurs, and this is also limited by the diameter of the shaft When once the reservoirs are filled, the bearings will run for months

without attention, although this would not be neces sary, as the oiler, or millwright, would attend to thi regularly. The hanga in Fig. 18 has an o level in the form of glass tube fitted. s that a correct supply i always available. Th top oil groove and th oiling rings can also b seen. The bearing i adjustable for heigh by means of the top an bottom support arms which are screwed an fitted with a lockin screw. The bearing ca also pivot around th supports so that align ment is fairly easy. The hangar fitted to the counter shaft in Fig. 20 is mounted in spherical pads, whic

FIG. 18. ADJUSTABLE HANGAR

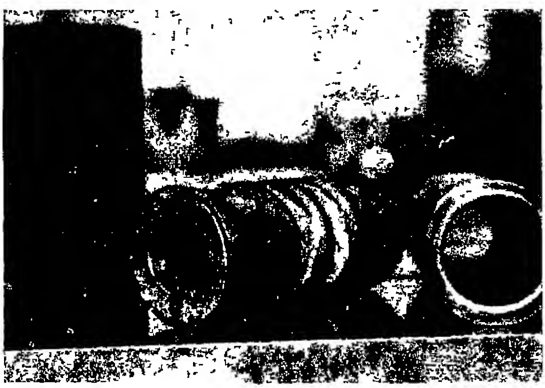


FIG 19. MOTOR BEARING, WHITE METAL LINED

Fig. 20. Hanger with Spherical Seating and Oil Catcher

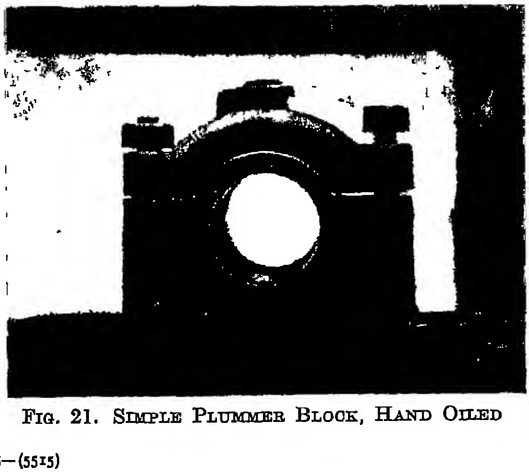


Fig. 21. Simple Plummer Block, Hand Oiled

guarantees that the shaft is not bound in any way, and ensures longer life to the bearings.

On elaborate and heavy shafting of large diameter small separate pumps are used, driven from the shafting to supply the lubricant and ensure a perfect supply. In other special cases, the bearing itself is water-cooled

FIG. 22. PLUMMER BLOCK, RING OILED

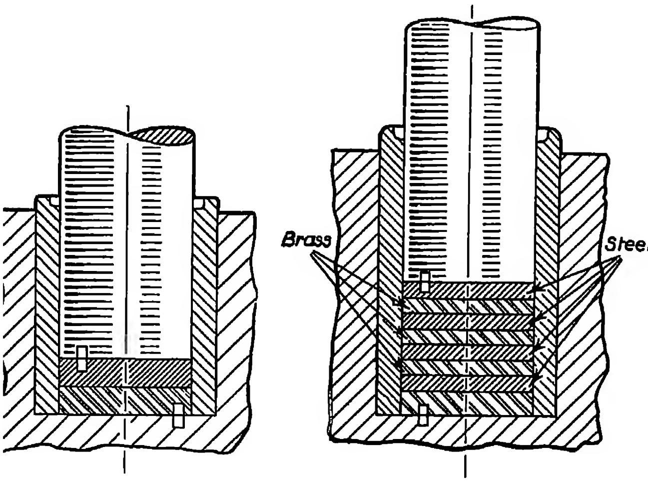
by means of inserted pipes through which a flow of water is maintained.

The details of standard plummer blocks (Fig. 21) are sometimes similar to hangars and the brasses are pegged in, or in older types the outside section of the brasses are square, hexagon, or octagon, and fit into correspondingly shaped housings. Where white metal bearings are used they should be held securely in the housings, either by pegging, grooving, or serrating. The oiling arrangement in Fig. 21 is only by hand, but the one in Fig. 22 is ring oiled. Notice the two hexagon plugs on the bottom half, one at the side for the oil level, and the one at the base for draining purposes.

STEP BEARINGS

With shafting and hangars, there is hardly any thrust to trouble about, but if the shaft is vertical, there is

he weight of the shafting and the pulleys to support. n the simpler type of step bearing, a pair of thrust vashers are used, one fixed to the end of the shaft and



FIGS. 23 AND 24. SINGLE AND MULTI-WASHER FOOTSTEP BEARINGS

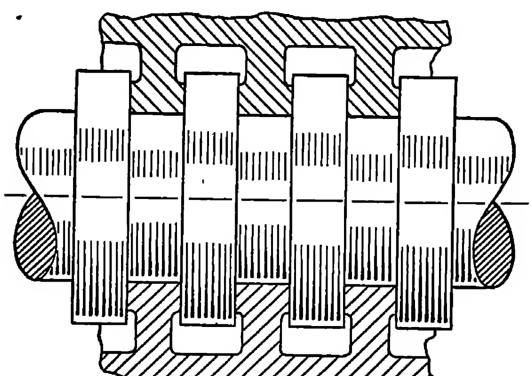


FIG. 25. COLLAR TYPE BEARING

the other fixed in the bearing housing. In a more important case, a number of discs are used, and the discs are alternately hardened steel and bronze, the end ones being fixed as before.

COLLAR TYPE BEARINGS

With a shaft of large diameter, a step bearing is unsatisfactory, as the velocities and, therefore, the wear varies at different points. For instance, imagine a shaft nearly 12 in. diameter, the velocity, say, at 200 revs. per min. would vary from nil at the centre of the shaft to 300 ft. per min. half way, and 600 ft. per min. at the outside diameter. The larger part of the centre is therefore cut away and only an outer ring used, but a much better way is to use a collar type bearing. As shown diagrammatically, a number of collars are turned on the shaft, and these bear up against similar pieces in the box, in this way the extra bearing is obtained, not by a larger diameter, but by increasing the number of bearing surfaces.

Vertical bearings such as used on water turbines, pumps, and vertical generators are the most difficult to lubricate, as the oil runs away by gravitation. The best means to use, therefore, is an oil groove cut either right or left hand according to the rotation of the shaft, which when revolved will act on the principle of an archimedean screw, and carry the oil up the shaft to the bearing.

MACHINE TOOL BEARINGS

On precision machines, and especially grinding machines, great care must be taken to ensure that no dust or fine abrasive can obtain access to the bearings, as otherwise wear and the consequent bad finish inaccurate work, and also chatter marks will be produced. With internal grinding machines, the spindles

so have to run at extremely high speed, and therefore
eed careful adjustment. The bearings are usually
pered, and also screwed each end for adjusting rings.

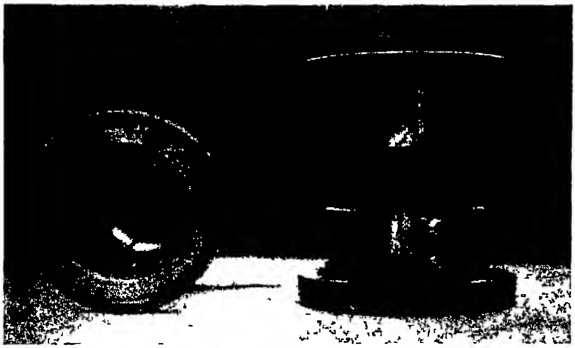


FIG 26 ADJUSTABLE TAPER BEARING

FIG 27. SPINDLE WITH TWO ADJUSTABLE TAPER BEARINGS

he taper is used for adjusting the size of the hole, as
ie bearing is forced in and out of the housing by means
f the screwed rings at either end. A bearing with the
ngs on is shown in Fig. 26, and a complete dismantled
pindle in Fig. 27. As will be seen, there is a taper at

both ends of the spindle, and a pair of adjustable bearings are fitted. These are screwed into the housing until all end-play is removed, and only running clearance allowed. With taper bearings as described, the thrust is not taken by the taper which would cause it to bind, but by a separate thrust washer ; the taper is

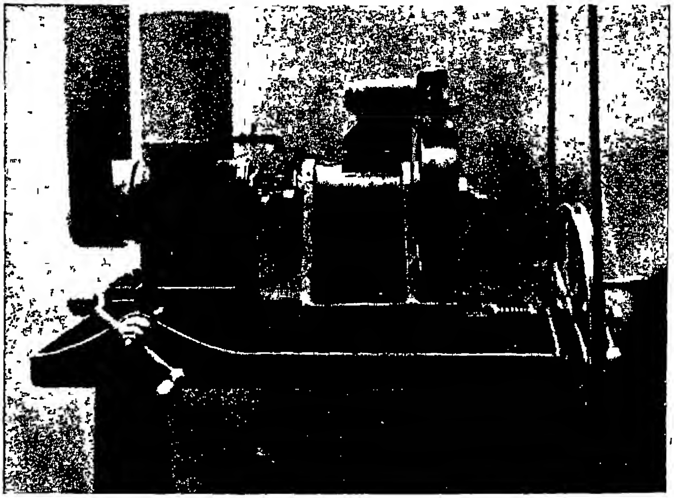


FIG. 28 SPECIAL MACHINE WITH ROLLER BEARINGS

only for adjusting running clearance, instead of having to strip down and file faces as in the standard two-piece type of bearing. Whatever machine or engine is used, it should be kept as clean as possible as there is less chance of foreign matter getting to the bearing or in the lubricant. In Fig 28 a special machine is shown in which roller bearings are used throughout, except for the slides, which are supplied with a greaser and also felts to protect the surfaces. On the drilling machine spindle which only has vertical movement, a

ain bearing is used, but is split so that all play n be taken up. This is also the case with the press indle which is of square section. Notice the means

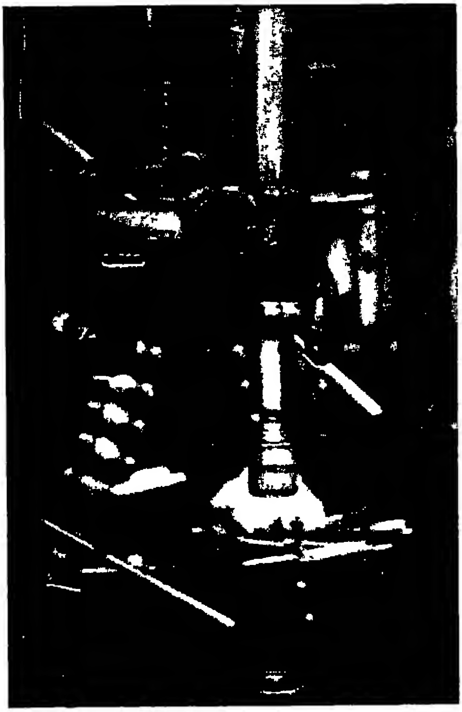


FIG. 29. DRILLING MACHINE WITH SPLIT BEARING

adjustment with the two bolts pulling up against the d of the adjusting pin.

Although bearings are usually split across the centre, is is often only because of economical production. Fig. 31, the main bearings for a compressor are ustrated, and it will be noticed that the splitting is

FIG. 30 PRESS WITH SQUARE SECTION BEARING

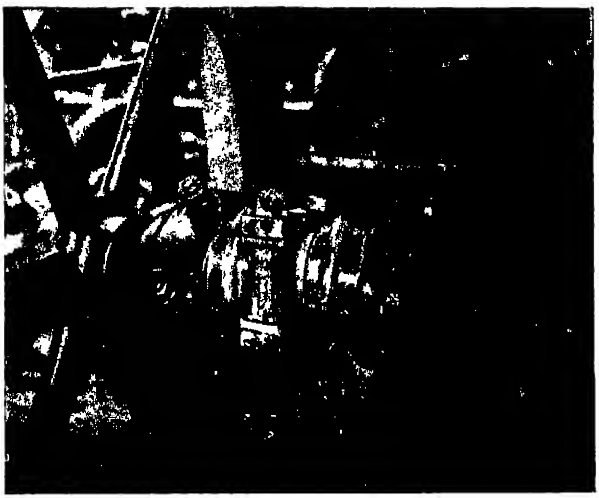


FIG 31 BEARINGS OF AN AIR COMPRESSOR

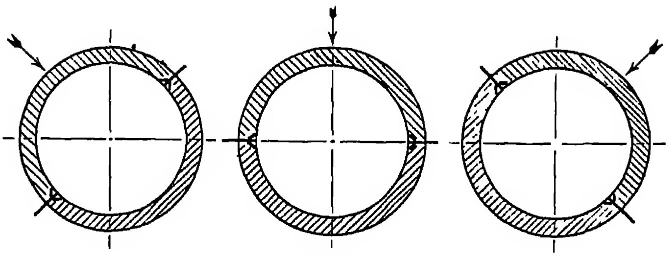


FIG. 32. LINE DRAWING OF SPLIT BEARINGS

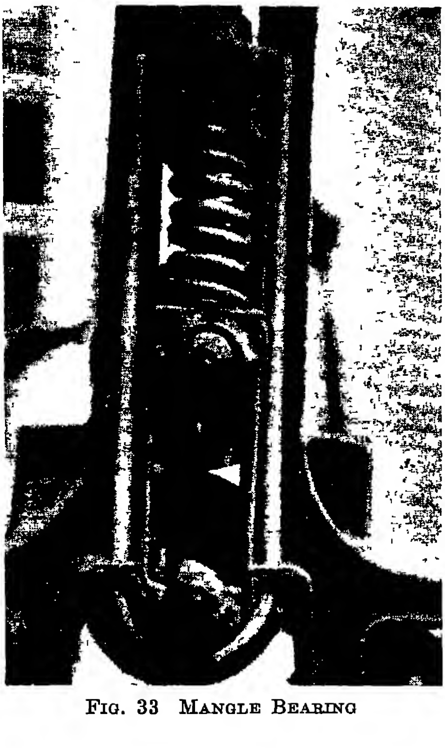


FIG. 33 MANGLE BEARING

at an angle or at the point of least pressure. The drawings (Fig. 32) indicate the line of pressure, and how the splitting of bearings should take place. Neither is it necessary to have a complete bearing. For instance, in locomotive wagon axles and also such household things as mangles, only half bearings are used as the pressure is only one way, and the other half if fitted would only be waste metal. A mangle bearing is illustrated in

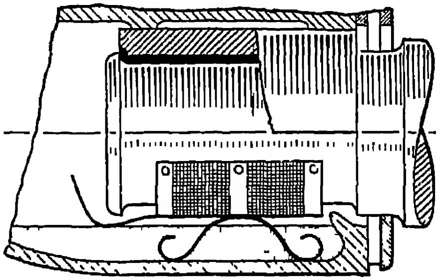


FIG. 34. DRAWING OF LOCOMOTIVE WAGON AXLE BEARING

Fig. 33, and Fig. 34 is a line drawing of a bearing as fitted to locomotive wagons.

The bearing only fits on the top half, and the bottom half has an oiling pad held in contact by a spring. Other types are lubrication by means of cotton waste and grease. The journal of the axles are rolled to obtain a good finish. A plain roller is mounted in the tool-box of the lathe and forced against the journal, which slightly compresses the metal and produces a hard surface. Knife edges are another type of bearing, although in an entirely different class. They are much used on weighing and balancing machines, where the least possible amount of friction can be allowed.

AUTOMOBILE BEARINGS

In motor-cars we have the main engine bearings of the journal type, gudgeon pin bearings which only have

intermittent action, and cylinder bores which have sliding action only. The materials vary also, white metal for main bearings, phosphor bronze for gudgeon pins, cast iron for cylinder bores, oilless bushes for

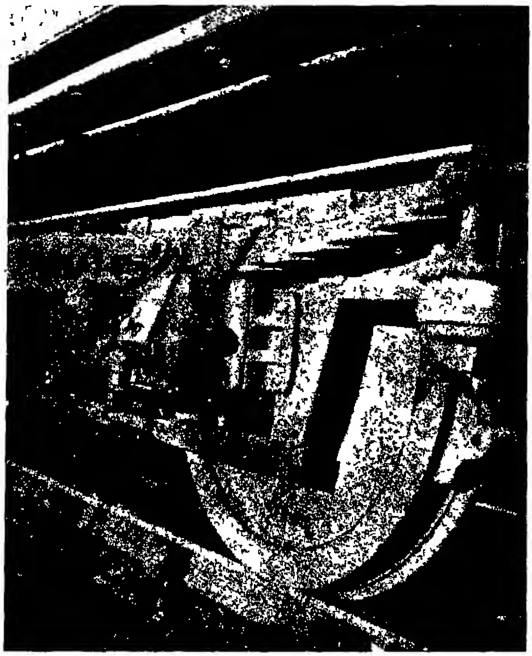


FIG 35. AXLE BEARING ON LOCOMOTIVE WAGON

hackles, stainless steel for such parts as brake cam-hafts, wood bushes for shock absorbers, and fibre ushes for contact-breaker pivots. Where a small mount of wear only takes place, solid one-piece ushes are used, but often these have to be split so hat the parts can be assembled. The big end and nain crank journals are mostly of white metal in a

gunmetal shell, although a few makers are now fitting roller bearings in special cases. The first two figures

FIG. 36. WHITE METAL LINED BEARINGS FOR PETROL ENGINE

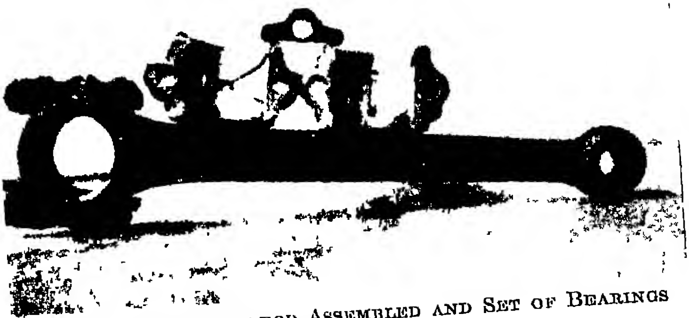


FIG. 37 CONNECTING-ROD ASSEMBLED AND SET OF BEARINGS

illustrate a set of main bearings, and a connecting rod with a set of bearings and the packing shim. In all cases the oil hole feed is in the top half, but it will be

noticed that the oil distributing grooves are in the top half of main bearings, and in the bottom half of the connecting rod bearings. This is in accordance with the idea that oil feeding grooves should be at point of least pressure. When the engine is driving the car, the load is delivered by the connecting rod being forced

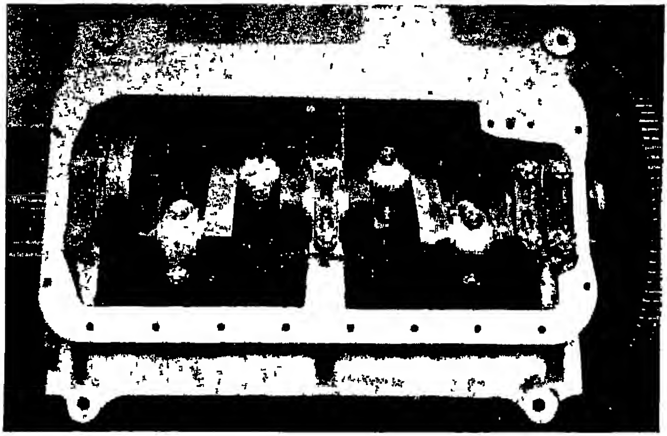
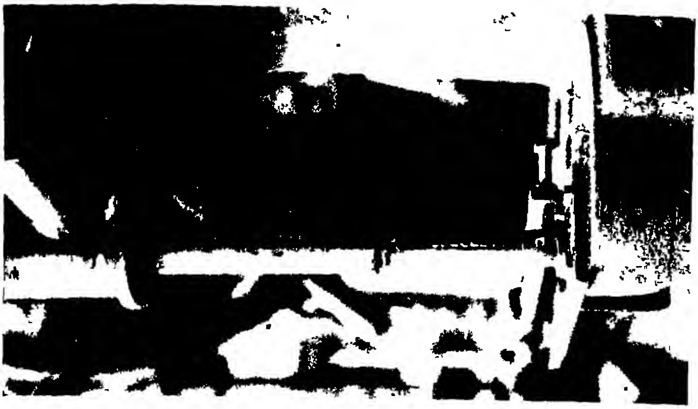


FIG 38 UNDERNEATH VIEW OF ENGINE SHOWING CRANKSHAFT ASSEMBLY

down, and this in turn transmits the power to the crank, therefore the pressure is on the top half of the rod and the bottom half of the main bearings. A general view of main bearings, and also the connecting rod bearings, is given in Fig 38 It is the underside of a 12 h.p. engine. The main bearings are very strongly webbed, and also have steel straps to strengthen further the case.

Although the gudgeon pin bearing in racing engines is force fed, the majority rely upon splash feed, or use a scraper ring, which removes surplus oil from the cylinder wall and delivers it to the bearing.



BALANCE OF REVOLVING PARTS

As engines now run at very high speeds, it is necessary to balance the rotating parts so as to ease the

FIG 41 SHOWING MAXIMUM GRAVITATIONAL MOVEMENT OF CONNECTING ROD

work of the bearings. The reciprocating parts, such as pistons and connecting rods, are also balanced, all of which tends to produce a smoother running engine. The illustrations opposite show how the small end of the

rod is finished to size, and also the oil grooves broached. When broaching the hole, the last few teeth on the tool do not cut but only burnish. They are of round section and slightly compress the metal, giving it a hard surface which will resist wear, and also impart a good finish. The broaching is done on a small hand press as only a light force is required. The fitting of bearings will be considered in a further chapter, but

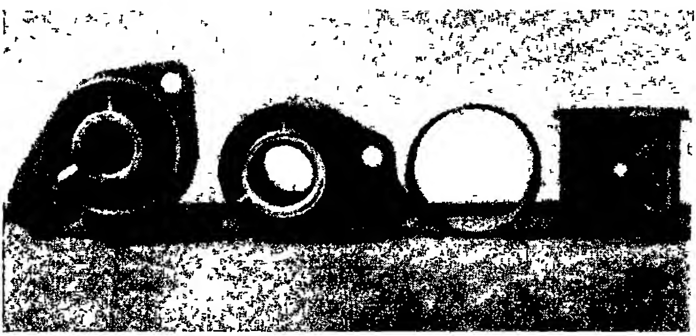


FIG. 42 GROUP OF WHITE METAL DIE-CAST BEARINGS

the illustration (Fig. 41) shows the movement allowed for a connecting rod to swing by gravitation. If properly fitted, run in, and, of course, lubricated, the rod by its own weight should fall from the top to the bottom position, where it is shown as a ghost outline Fig. 42 shows a few bearings which are die castings of white metal without any shells of gunmetals. They are perfectly satisfactory as long as there is no pounding on the bearing, but they cannot carry heavy loads as the white metal will spread.

BEARINGS USED ON SHIPS AND TURBINES

The engine and driving mechanism of ships are a study in themselves There are the usual types of

bearings in the engines, but the bearings for the propeller shaft to take the load and also the thrust are exceptional. In the old ships the bushes in the stern

FIG 43. SHIP'S PROPELLER SHAFT BEARING

ube were made of a wood (lignum-vitae), as it was n contact with the water, and this was the best material hen known. The water on lignum-vitae acts as a ubricant, and being a hard wood it was satisfactory. This material is now superseded by metal, and special rrangements made for lubrication. On small boats

there is the standard type of greaser as shown in Fig. 43, but in large vessels the lubrication is carried out by pumps, and the apparatus is accessible from inside the ship. Lignum-vitae is still used on hydraulic machines, pumps, and vertical turbines. Fig. 44 illustrates a large journal bearing used on a ship. It is

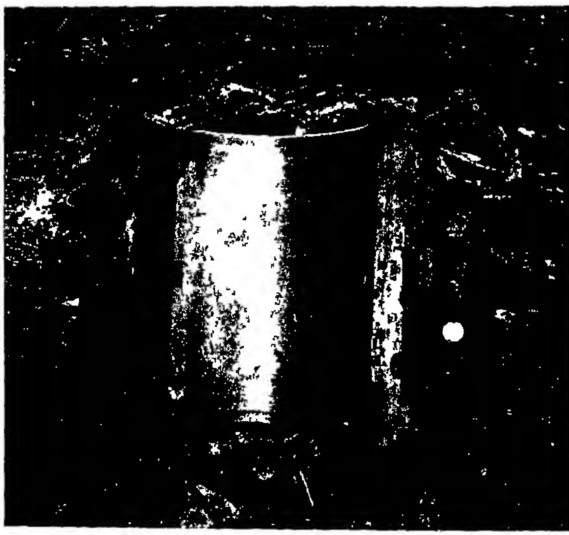


FIG 44. VIEW OF LARGE JOURNAL BEARING

over 12 in. in diameter, and, of course, the material is gunmetal lined with white metal The bearing is in the cap, and it will also be noticed that the oil delivery groove is at the side, this being the point of least pressure, and connected up to the oil supply channel. With important bearings a safety strip of bronze is put slightly below the surface, which would support the spindle in case of the white metal running. The bearings of turbines are often provided with spherical seatings

THRUST BEARINGS

Owing to the enormous thrust from the propeller, pecial attention has to be given to the bearings to take his load. Single thrusts are useless, and the collar type vas the recognized medium, being brought to a high tandard for this kind of work. The collars were made n separate halves and inserted into the shaft, which .llowed special hardened material to be used, and the earing pieces were all made adjustable so that each

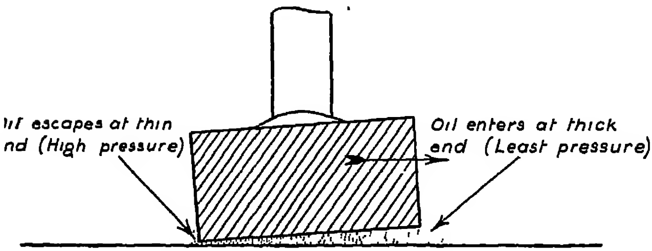


FIG 45. DIAGRAM SHOWING WEDGE-SHAPED OIL FILM

ook its share of the load The standard type was of a orseshoe shape, so that any single thrust could be aken out for inspection. In another type of collar earing, thrust was provided for forward and reverse, ie top half of the bearing acted forward, and the ottom half reverse. These could be adjusted so that iere was no play in the spindle. The greatest trouble as heating due to bad lubrication.

MICHELL BEARINGS

Of recent years, a great advance in the design of earings and lubricating has been made by the introduction of the "Michell" bearing. As will be dis-ussed in further paragraphs, a shaft in a bearing does

not run concentric, as the oil film spreads and forms a taper wedge, commencing at the point of least pressure and gradually tapering off as shown in the drawing, Fig. 45. The Michell bearing takes advantage of this fact, known as Reynolds' theory, and consists of pads which are pivoted, so that they can swing over and allow the oil film to form the taper wedge. The pads are of approximately square section, but they are modified to suit

FIG. 46. MICHELL THRUST BEARING PADS

different conditions; for instance, the large pads in Fig. 46, each about 12 in. across, are for taking a thrust, and shaped accordingly to suit the flange. One is placed face downwards so that the spherical seat can be seen which allows the swivelling motion. There is no necessity for oil grooves of any description, in fact, they would be a disadvantage owing to upsetting the oil film. The Michell bearings are designed for use either as journal or thrust bearings. To show the great advance made, a few figures will be given. With a collar type of bearing only a bearing pressure of 60 lb. per

sq. in. could be satisfactorily used, but with the advent of Michell bearings, 500 lb. per sq. in. at speeds up to 100 ft. per sec. is standard, and tests have been carried out in which a pressure of 3000 lb. per sq. in. has been sustained for continued periods, and even loads of 5 tons per sq. in. have been imposed without failure.

FIG. 47 MICHELL THRUST BEARING FOR TURBINES

A complete assembly of a Michell bearing as applied to a turbine rotor, showing the thrust pads in position and a loose one above, is given in Fig. 47. In use the bearing runs in an oil bath, and is attached to the frame of the turbine.

Another advantage is that, due to their compact design, the length of the machines can be cut down considerably.

LUBRICANTS

There are hundreds of different oils for lubricating purposes on the market, but they can nearly all be divided into a few divisions. namely, mineral oils, vegetable oils, animal oils, and compounds, the terms giving the source of the products. Mineral oils are derived from such typical substances as oil shales, and are supplied from the distillation of crude petroleum and vegetable oils, being obtained from plants such as the castor oil and olive oil. The animal oils are well known under such names as sperm oil from the whale, and neatsfoot oil from the feet of beasts. The compound oils are a mixture of any of those already mentioned. Owing to the huge demand for petrol, it follows that the bulk of oil belongs to the class known as mineral oil. There are other lubricants, such as graphite and grease, and also water, which is a lubricant for lignum-vitae. The lubricants have differing physical properties which are useful for special cases One of the most important properties is viscosity, or (a simpler, even if not quite correct, term) thickness. Water or paraffin has a low viscosity, and oils, grease and treacle have higher figures. The greater the viscosity, the greater the carrying load can be, as it is harder to squeeze the thicker lubricant out, but it must also be remembered that it will lower the speed. A rough rule is that the larger the bearing the heavier should be the oil, and that as the speed increases, so should a thinner oil be used. The light oils should only be used with a light unit bearing pressure With any new machine, the oil should be changed after a short time, as it becomes contaminated with the sand, etc., which cannot all be removed from the castings. With machines which are affected by the weather, such as motor-cars and cycles, it follows that during the winter a different oil is often

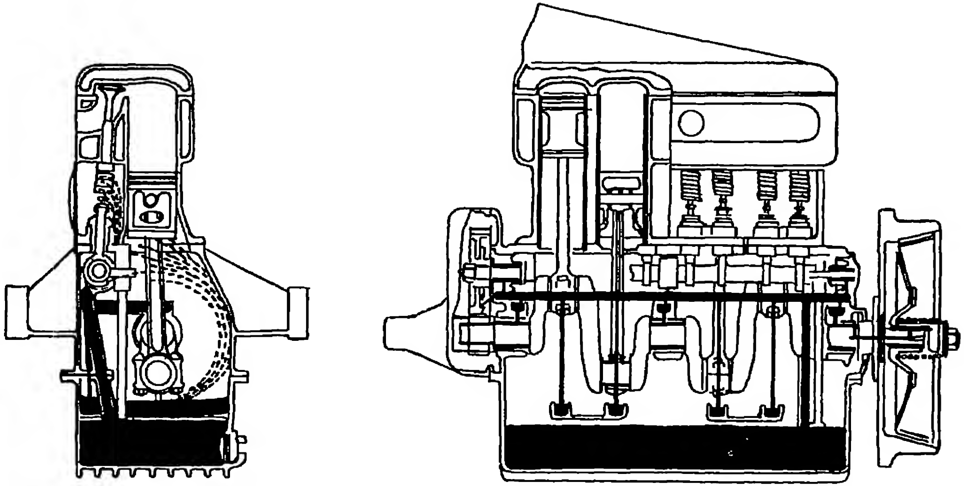


Fig. 48. Showing Typical Distribution of Oil to the Bearings of a Petrol Engine

necessary. The various makers usually specify the type in their instruction manuals, and these should be carefully studied. Another rule is, never mix oils, as when different oils are mixed a sludge or deposit is likely to occur and foul the complete machine, or in any case will produce hot bearings. In some compressors rubber packings are used, and as mineral oil destroys rubber, the lubricant which should be used is glycerine. Lubricants need not be in liquid form, as besides thick grease, there are still graphite, talc, etc., which although often mixed with a liquid are, however, sometimes used quite dry in powder form. Examples are lace-making machines, chocolate machines, etc., and the powder is either dusted on or syringed into the bearing. Graphite in an extremely fine condition and mixed with oil is a good lubricant, and imparts a fine finish to bearings. It has been found that if two unlubricated surfaces are moved, the amount of force required is proportional to the weight of the load acting on them, but if the surfaces are correctly lubricated, the amount of force is very greatly reduced, so that much heavier loads can be carried; in fact, the limit is the viscosity of the lubricant. The action of the lubricant can be likened to that of ball bearings, the atoms of the oil sliding over each other and not allowing metallic contact. When bearings seize, the reason is because of metallic contact due to the breaking of the oil film; in fact, if a perfect film of oil could be guaranteed in the bearing, the material of the bearing would not matter. The oil film does not always work, although this is the object aimed for, and another property called oiliness comes into play. It has been noticed that with two different oils having similar properties, one was more satisfactory as it appeared to coat the surfaces where the other failed; in fact, this is a great feature with the castor oil.

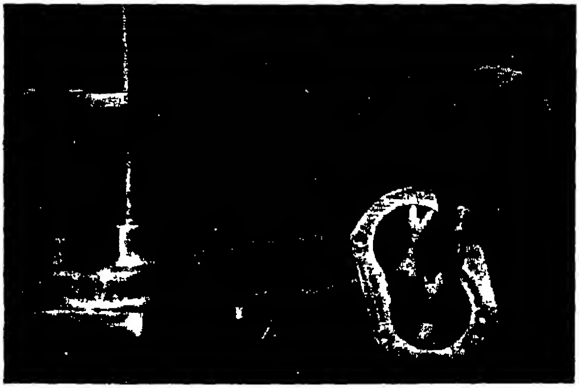


FIG. 49. GEAR TYPE OIL PUMP WITH COVER REMOVED

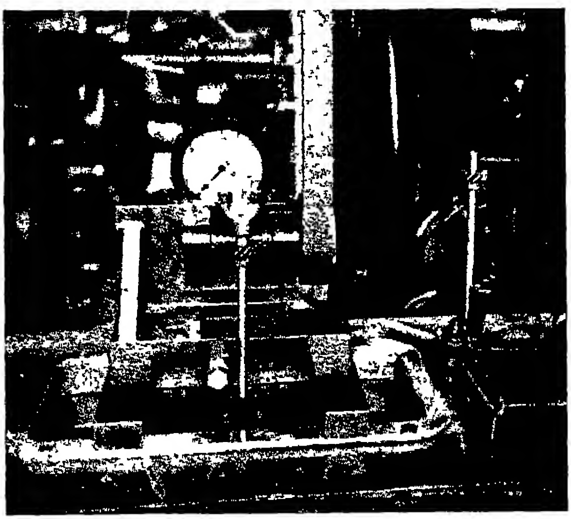


FIG. 50. TESTING STAND FOR OIL PUMPS

LOW AND HIGH PRESSURE SYSTEMS

The lubricant is sometimes fed into the bearings by gravitation alone, but more often is pumped in under pressures, varying from a pound or two to hundreds of pounds per square inch, according to each special case. The lubricating of motor-cars has been specially studied, and every part of the engine details is automatically lubricated under pressure, and a few firms carry out a similar system with the complete chassis. Fig 48 illustrates the main lubricating parts of a special engine. The oil is contained in the oil sump which has cooling fins cast on, and a pump is submerged in the oil so that no pressing is necessary. The lubricant is pumped up the pipe to the main distributing pipe running the whole length of the engine, and from this is fed to the main crankshaft bearings, the camshaft bearings, timing gears, clutch spigot, and also to troughs into which the connecting rods dip and fling the oil to the cylinder walls. With pressure systems an oil release valve is necessary, and it is usual to have this adjustable so that the oil pressure can be easily regulated.

OIL PUMPS

The gear pump is the type which is most used, and Fig. 49 gives an outside view, and also a view with the bottom cover removed to show the gear wheels. As the gears revolve, they suck the oil through the inlet pipe and deliver it to the opposite side, where it is forced up the outlet pipe. By fitting ball valves or similar apparatus, they can be made to operate in both directions. Backlash in the gears does not matter, in fact, a small amount is an asset, and a fairly high efficiency is obtained.

Fig 50 shows a test being carried out on a gear pump, where the following figures were obtained which

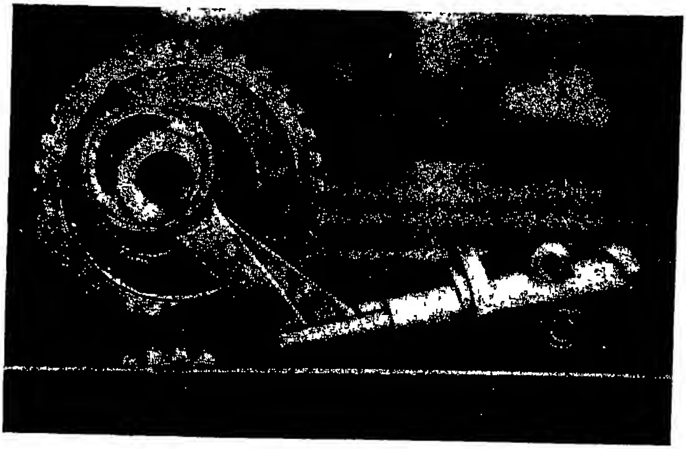


FIG. 51. PLUNGER TYPE OIL PUMP

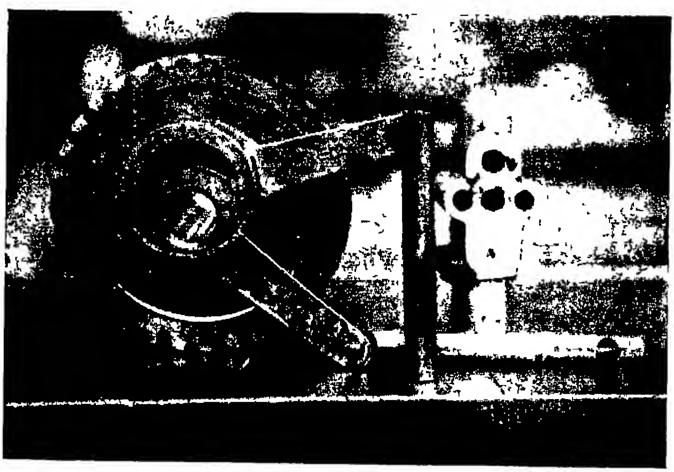


FIG. 52. ANOTHER VIEW OF PLUNGER TYPE PUMP

indicate the drop in pressure when the temperature is altered. At 50 degrees the pressure was 80 lb. per sq. in., but at 80 degrees the pressure dropped to 20 lb. Two other types of pump are the rotary pump and plunger pump. Two views of a plunger type are shown in Figs. 51 and 52. In this case there are two plungers driven by eccentrics from the gear wheel. The front is the plunger proper, and the rear one acts as the valve, which thereby gives a pump with no valves, or loose balls to get out of order.

Before leaving the question of lubricating, it is well to point out the importance of oil strainers or filters. They should be fitted in all cases and should be accessible. With such parts as locomotive axles where cotton waste soaked in oil is used, this itself feeds the lubricant, and also excludes dust, dirt, and grit, but with force feed any dirt present is also forced to the bearing unless first removed by a filter.

Undoubtedly, the perfect way to lubricate is to have a supply of oil entering the bearing, so that the film is not broken where the pressure is greatest. When this takes place the journal of the shaft will be eccentric to the bearing, the oil entering at the thick edge and leaving at the narrow edge, as explained in the chapter on Michell bearings. While we have not space to go into the question of oil purifiers and rectifiers, it is as well to point out that these are now being fitted to high grade motor-cars, and, undoubtedly, it is a step in the right direction.

OILERS (HAND)

The simplest and best known type of hand oiler is the oilcan, and there is not much variation in the oil cups except the size. Some are only receptacles, while others have a wick feed which filters the oil and only allows a small quantity as required. In Fig. 54 a

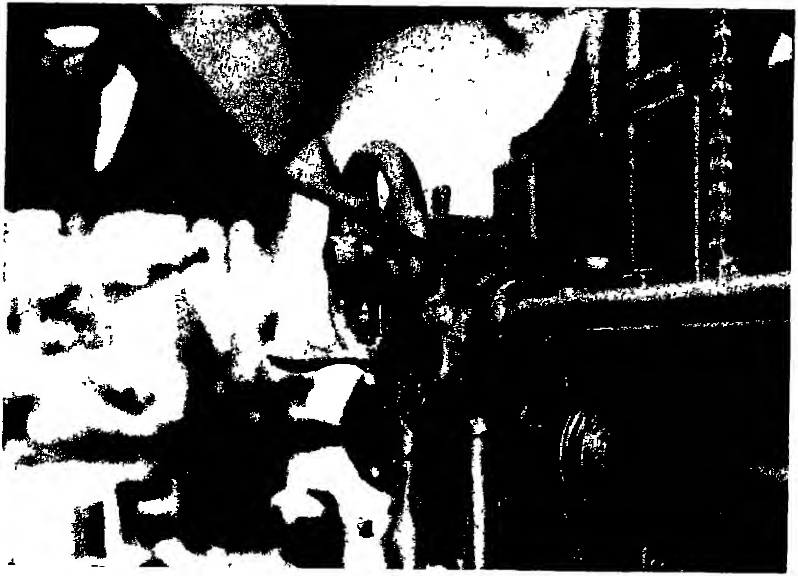


Fig. 53. Hand Oiling

wick-feed type is illustrated. The wick is fed from th reservoir and passes through the cap, and then dowı over the bearing, which is only a half bearing. Th amount of lubricant is under control, as the cap can b adjusted to press tightly on the wick which reduces th supply. The standard type of oilcan, and also a com mon form of oiler, is shown in Fig. 53. There are many

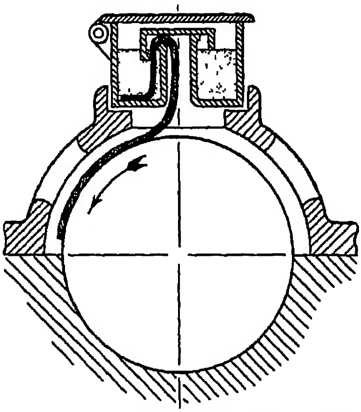


FIG. 54. LUBRICATION BY WICK AND RESERVOIRS

types, some with spring cap, sliding cover, etc Another device is the felt pad which will hold a supply of oil, and feeds it on to the bearing. A useful type of oiler for small work is the spring-well, in which the inlet is kept closed by a ball and spring until the oilcan spout is forced in.

OILERS (MECHANICAL)

Under the class of mechanical oilers come pumps, etc., which we have already discussed, and also such things as oiling rollers. The ways of machine tables must be lubricated, especially in such cases as planers and grinders which are in use continuously. This is

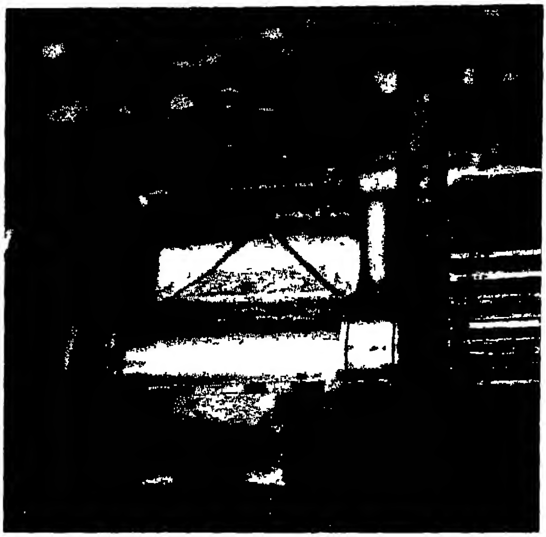


FIG 55. OILING ROLLERS ON GRINDING MACHINE

FIG. 56. OILING ROLLERS ON PLANING MACHINE

FIG. 57. STANDARD TYPE GREASE CUP

FIG. 58. GREASING BY MEANS OF GREASE GUN

done by rollers, and two figures show how these are sunk in oil wells and mounted on a light spring, so that they are always in contact with the table. Whenever the table moves, it turns the rollers which carry oil to the ways and deposits a lubricating film.

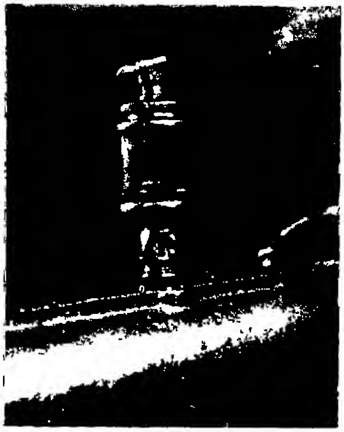


FIG. 59. DRIP FEED LUBRICATION SYSTEM

GREASERS AND GREASE GUNS

With greasers we have a larger variety. There are the usual grease cups with screw-down caps, which are operated when a supply is required, and another type in which the cap carries a spring-loaded plunger, which forces the grease into the bearing as soon as there is space for it. A type of greaser used extensively on motor-cars is the nipple type, in which a grease gun is used to force the lubricant into the bearing (see Fig. 58). The nipple only has a small hole, so that water, etc., cannot obtain access, and the grease gun end is coned so that no screwed connections are necessary.

DRIP FEEDS (SINGLE)

With more important bearings, where it is necessary that the amount of lubricant used can be verified, sight feeds are used. A standard type is illustrated in Fig. 59,

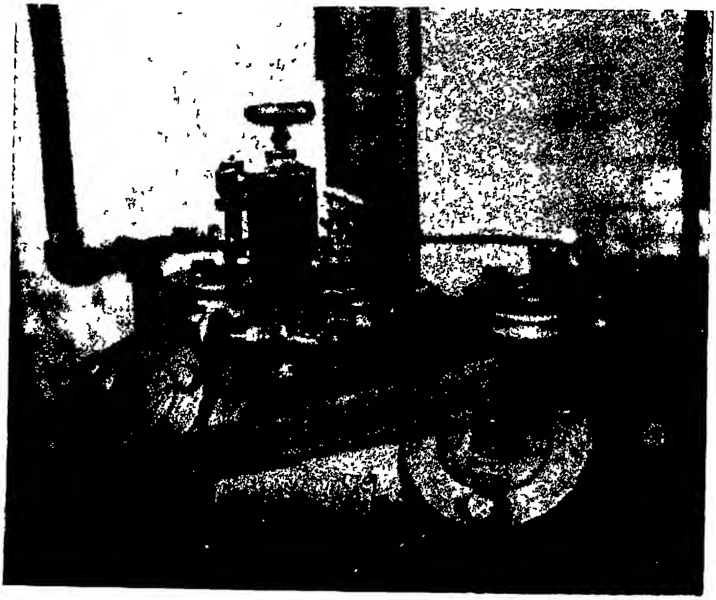


FIG. 60. DRIP FEED FITTED TO COMPRESSOR

and consists of a glass chamber to hold the oil and a needle valve at the bottom. The oil drips from the outlet, and can be seen and regulated by the small milled screw at the top. With the small handle at the top in the horizontal position as in figure, the needle valve is closed and no oil passes. To re-commence delivery, the handle is placed vertical Another type of drip feed is illustrated in Fig. 60, and is fitted on a

ompressor to feed into the cylinder. With the ordinary ype the oil would be blown straight back with the ressure from the compressor, and therefore it is neces- ary to fit an equalizing pipe, so that the pressure in

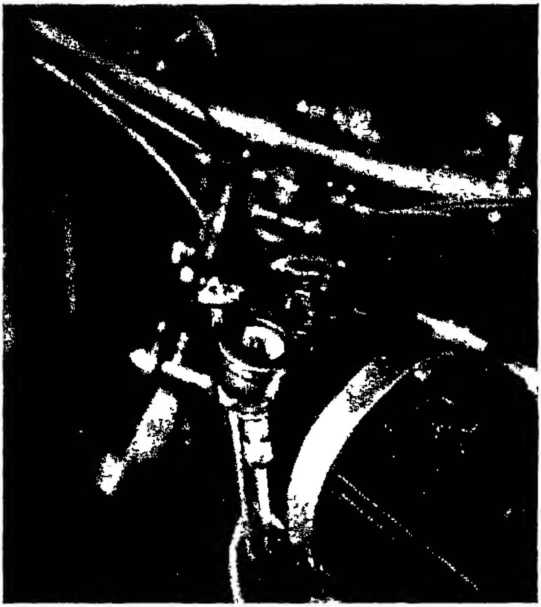


FIG. 61 SIGHT FEED AND PUMP ON A MOTOR-CYCLE

e drip feed is the same as in the cylinder and the can then flow in The type most known to readers l be similar to that in Fig 61, as fitted to many kes of motor-cycles for lubricating the engine. ere is a plunger pump, the discharge of which is ially operated by a spring. This pumps the oil ough the drip feed, and the amount can be regulated a milled adjusting screw, and, of course, checked

by watching the amount through the glass sight feed.

DRIP FEEDS (MULTI)

Figs. 62 and 63 indicate multi-drip feeds all fed from a common reservoir, one having three leads and the other twelve leads to different bearings on the machine. By this device, one filling and one tap controls the supply to all the bearings.

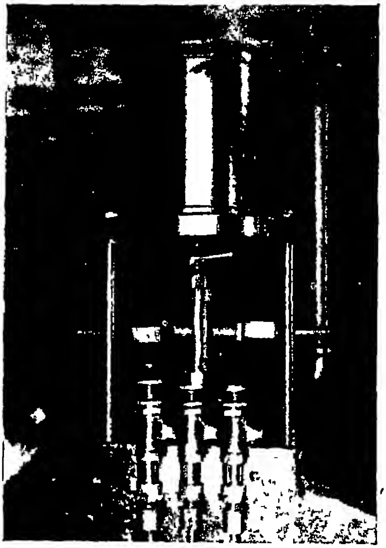


Fig. 62. Multi-drip Feed

RING OILERS

Ring oilers are extensively used in machine tool work, and have already been described under bearings for shafting. The reader is recommended to refer back to the paragraph dealing with this matter. The shape of the rings varies, as does also their weight, both of which are modified to suit the type of bearing. In one type of ring a channel section is used, which tends to bring up more lubricant than the plain type. The rings of oilers are usually about twice the diameter of the shaft. Another modification is the provision of scrapers and channels in the housing to take the oil from the ring and distribute to the oil groove. Instead of solid rings, chains can be used to deliver the lubricant, and Fig. 64 illustrates the

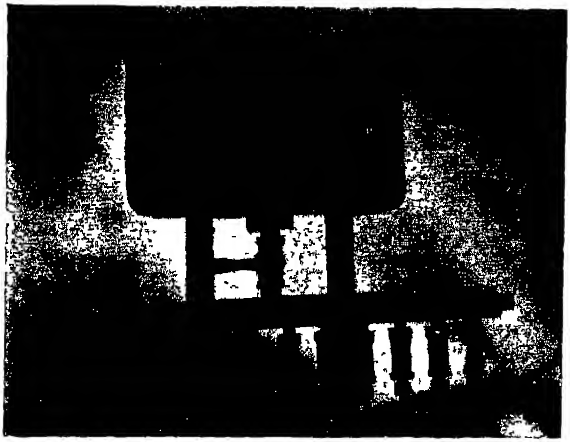


FIG. 63. MULTI-DRIP FEED, 12 LEADS

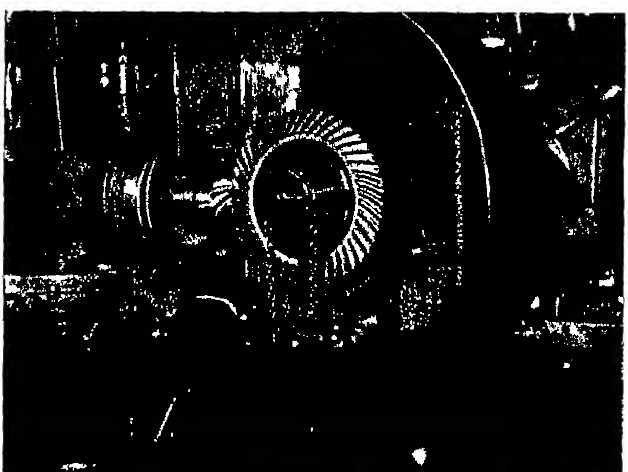


FIG. 64. LUBRICANT LIFTED BY CHAIN

use of a chain which lifts a mixture of oil and oilstone powder, and then by reason of centrifugal force delivers it to the teeth of the gears. At high speeds, chains are liable to churn up the oil and cause a foam

OIL INDICATORS

There are many types of oil-level indicators. Some, as shown in Figs 65, 66, have glass tubes in which the level of the oil is seen, while others have floats which move an indicator. There are simpler forms in which an indicating rod is used, as in Fig. 67, or again, cocks or taps are fixed at the various levels. Some bearings are also fitted with a thermometer to indicate the temperature, but in other cases the temperature of the oil is taken, so that any special increase is seen and steps taken to rectify the fault.

The materials of the component parts of bearings are extremely variable. The following are a few—

Mild steel and white metal bearing.
Cast iron and cast iron bearing.
Steel and cast iron bearing.
Hard steel and phosphor bronze or gunmetal.
Hard steel and agate.
Hard steel and fibre.
Soft steel and wood.
Stainless steel and brass.
Hard steel and chilled cast iron
Hard steel and hard steel.
Many varieties of oil-less bushes consisting of lead base, graphite, wax, etc., and also stone-lined bushes.

All of these have their special duties, which will be considered. There would be no need for such a huge variety of materials if a film of oil was always present, as then metallic contact could not occur. Another

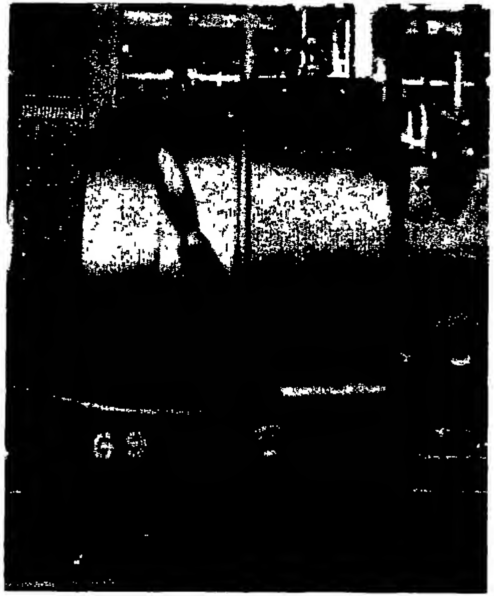


FIG. 65. LATHE HEADSTOCK FITTED WITH WINDOW TYPE OIL INDICATOR

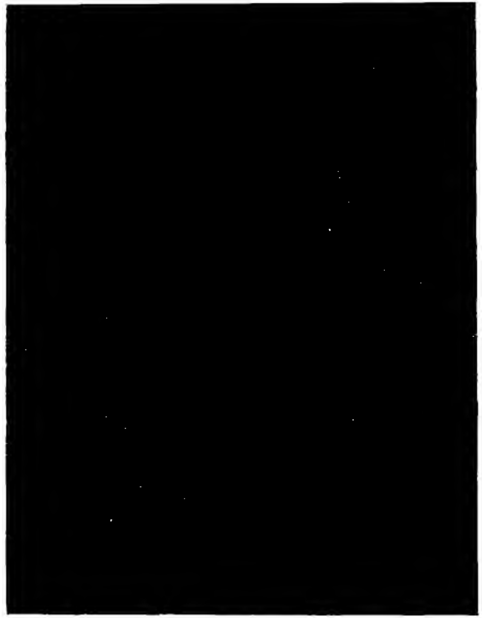


FIG. 66. FLOAT TYPE OIL INDICATOR

interesting point is, that it is not necessarily the composition or analysis of the material that is at fault, but it is the structure which is important. The best bearing metals are those having a mixture of hard and

FIG 67 SIMPLE DIP-STICK TYPE OIL INDICATOR

soft constituents distributed evenly throughout the material. Similar metals are more liable to seize than those of dissimilar material. Another reason for varying metals is the different lubricants, and also the bearing pressure, as these vary from 30 to 8000 lb. per sq. in. A hardened steel shaft in a hardened steel

bearing will stand a pressure of 2000 lb. per sq. in., mild steel in white metal, 500 lb. per sq. in., and cast iron in cast iron, 100 lb. per sq. in. These are only approximate, and would again vary according to lubrication, and also whether the load was constant or intermittent. The bearing area is equal to the diameter

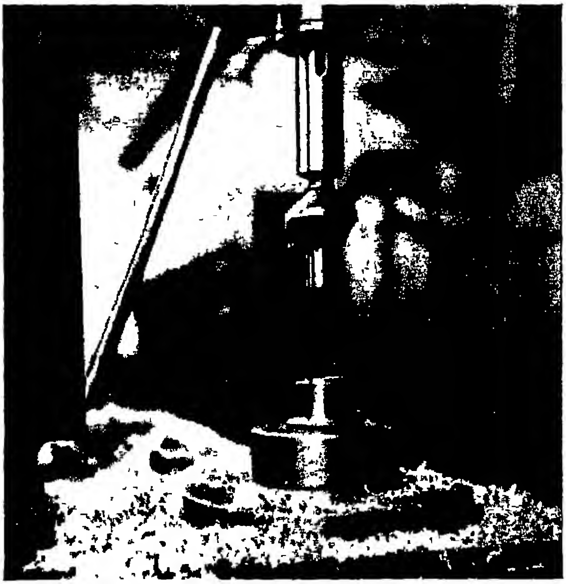


FIG 68 SHOWING THE HARDENED STEEL BEARING IN WHICH THE PEG OF A FACING CUTTER IS LOCATED

ultiplied by the length, and in motor-car engine prac-ce the bearing pressures vary from 500 to 1000 lb essure, according to whether splash or forced feed is ed for lubricating. For bearings such as gudgeon ns, these figures can be increased to 2000 lb. per . in. Bearing material must thereforo stand high

unit pressure at high rubbing speed without wearing excessively, and the friction must be low, or otherwise a lot of power is wasted. They must also resist corrosion, and the journal should be hardier than the bearing, so that replacement can be more easily performed. Hard steel bearings are extensively used on jigs and

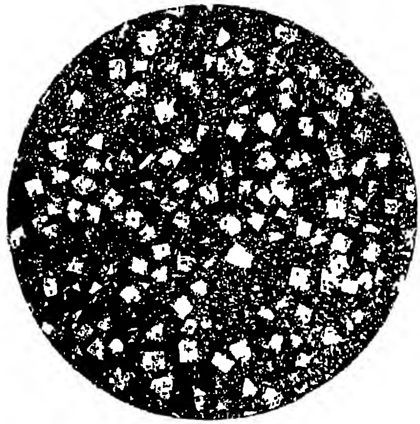


FIG 69 THE MICROSTRUCTURE OF A TYPICAL AUTOMOBILE WHITE METAL

tools, and Fig. 68 illustrates a bearing of this material where the peg of a facing cutter is located in the hardened steel bush which acts as a bearing during the cutting period.

WHITE METAL

The most important anti-friction bearing metal is the so-called white metal, or babbit. The composition varies, but is principally tin with antimony and a small percentage of copper. White metals, if examined microscopically (see Fig. 69), consist of cubic crystals of a hard substance embedded in a softer mass, and it is this arrangement which makes it so very useful.

The hard crystals are evenly distributed and take the wear of the bearings, and being in a softer mass they give, and in a very short time the surface is in a perfect condition. If subjected to impact blows, they must, however, be backed with a harder material. There is not much difference in the friction in the soft and hard metals, but with the softer metals, when run, they are soon reduced to a level and smooth surface which distributes the load over the bearing. With the white metals there is also less chance of scoring the shafts, as if overstressed the bearing will run hot and melt. Also if, through carelessness, grit happens to obtain access to the bearing, it will become embedded in the white metal, whereas if a harder bearing was fitted it would probably score the shaft

BRONZE

The bronzes are much harder than white metals, and much more work is required to produce a satisfactory bearing. The bearings must be properly bedded with the shaft or journal, as if not, the load is taken by the high points only, which will not give like white metal, with the result that heating occurs and seizure of the bearing is sure to follow, which will also probably score the journal Fig. 70 is a photograph of a headstock in which the bearing metal is bronze. This front bearing also takes the thrust through the steel washer, and the brasses are of square section and do not, therefore, require to be pegged

CAST IRON

Chilled cast iron is a hard wearing metal used in conjunction with a hard steel journal, but as with the bronzes, the bearing must be perfectly scraped or bedded in. If two bearings are required to be in line, such as a lathe headstock, trouble would be experienced

if both were hard bearings, and good practice is to have the front one of hard material, and the rear one of white metal which can be run in. The boring machine spindle in Fig. 71 illustrates the use of cast iron. Lead acts as a good bearing metal, and lead base bearings have given good service, but they will not stand such

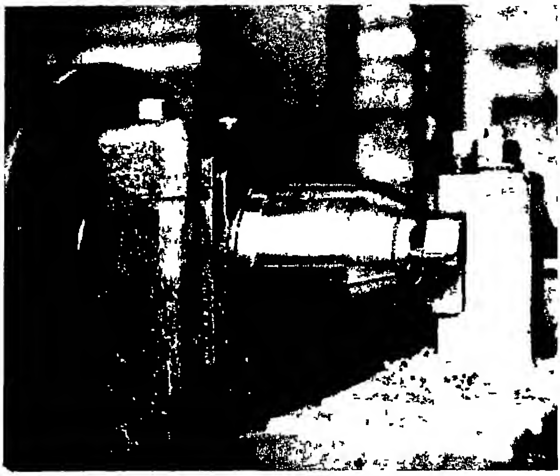


FIG 70 HEADSTOCK WITH BRONZE BEARING

a high compressive load as tin base bearings. The following is the composition of some railroad wagon bearings: 70 per cent lead; 15 per cent antimony; and 15 per cent tin. There is also a bronze bush on the market in which graphite to 40 per cent by volume is mixed in the metal. By reason of this the bush is porous, and will hold lubricant for some considerable time. Vulcanized rubber is the material which has been used for such parts as pumps and propeller shafts and, as with lignum-vitae, the water present in such cases acts as the lubricant.

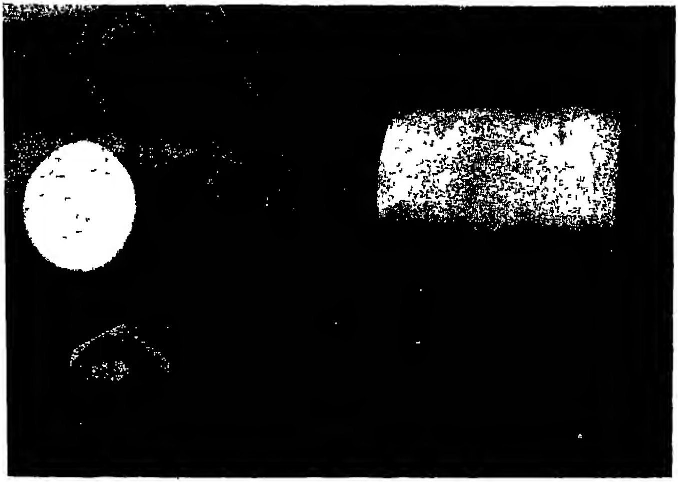


FIG 71 BORING MACHINE SPINDLE WITH CAST IRON BEARING

FIG. 72 PHOSPHOR-BRONZE CONNECTING-ROD BEARING

STONE BEARINGS

To hear of bearings made of stone will perhaps surprise a few of our readers, but stop to think of the watch in your pocket, or on your wrist, and remember that you had to pay for so many jewels which are really stones, or often agates are used. A banded agate in

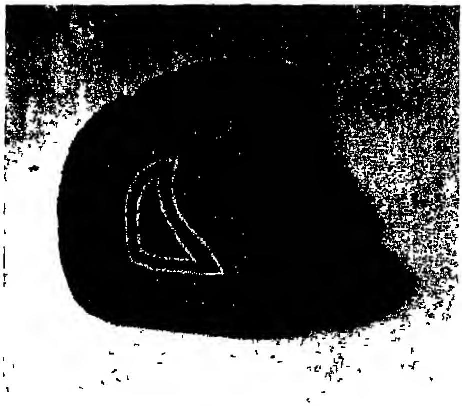


Fig. 73 Banded Agate in its Natural State

its natural state is illustrated, together with a photograph of jewelled watch bearings, in Figs. 73 and 74 Agates and precious stones are the hardest of minerals, and will therefore withstand wear. Ordinary tool steel will not cut them, and it is necessary to use their own powder, which acts as an abrasive, to machine them. A soft limestone has also been used in bearings, but for a different purpose. The bearing metal was white metal, and limestone slabs were inserted which, being porous, held a supply of lubricant, and was similar in action to a felt pad.

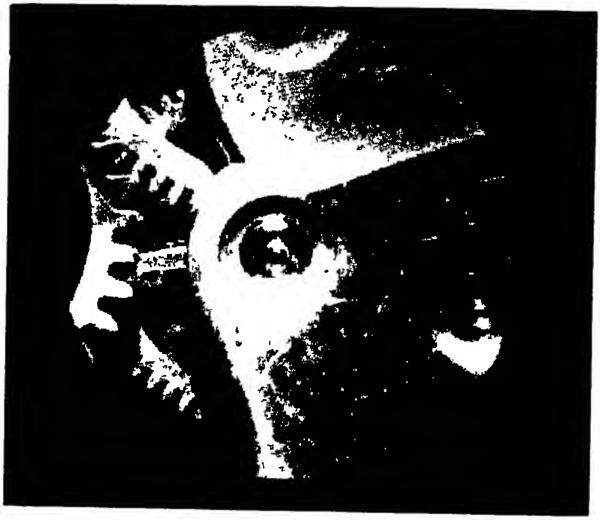


FIG. 74. TYPICAL JEWELLED WATCH BEARINGS

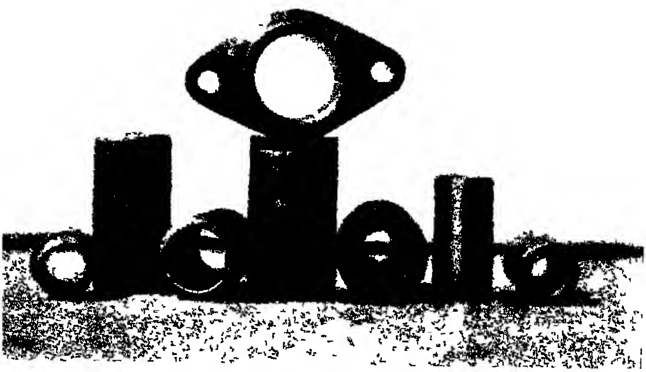


FIG. 75. GROUP OF OIL-LESS BEARINGS

OIL-LESS BUSHES

There are many different materials used in the manufacture of the so-called oil-less bushes, and photographs of some of the types are reproduced. In one case, wood is impregnated with wax or oil, and in others a bronze

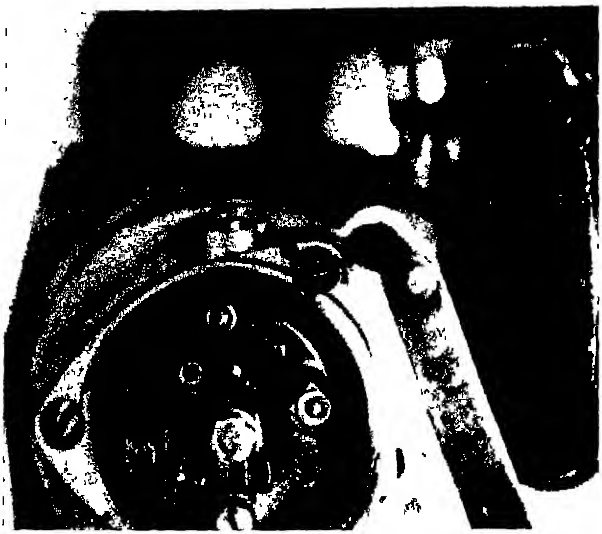


FIG 76. OIL-LESS FIBRE BUSH IN MAGNETO CONTACT-BREAKER

bearing is used with inserts of graphite. The lignum-vitae bushes do not require oil, as they naturally contain a resinous substance, and are usually used where water is abundant, which also acts as a lubricant. Lignum-vitae has been used mostly for the stern tube bearings of ships, and the propeller shafts were fitted with bronze liners on which the wood acted, so that no rusting could take place. Stainless steel is now being used, in conjunction with oil-less or bronze bushes, on

car parts which are not fully protected from the weather. With the oil-less bushes it often pays to add an oiler or greaser, and good examples are shown in

FIG. 77. WOOD BUSH FOR SHOCK ABSORBER

Fig. 78. These are shackle bushes for motor-cars and besides the bush, end plates are also fitted, and greaser nipples, so that a grease gun can be used occasionally.

FIG. 78 SHACKLE PIN BEARINGS

Oil-less bushes are also useful for such details as loose pulleys, textile machines, and specially in conveyors where the lubrication of bearings is apt to be forgotten

WEAR OF BEARINGS

The wearing of bearings is due to breakage of films, and through dirt and foreign matter obtain access. Most wear takes place at starting and l speeds, as when running at speed the shaft tends float. If a bearing is not correctly proportioned regards length, wear is liable to take place at the en as if so long the bearing will act as a beam and be When a bearing ru hot under stand conditions, an creased diameter do not help, as althou a larger surface obtained, the veloci also increases. To c tain a cooler beari increase the length, extra surface is o tained with no ext velocity The pa where most wear tak place is at the off-si of the bearings. This is the side of greatest pressure, a where the thin end of the oil film occurs (see Fig. 78A

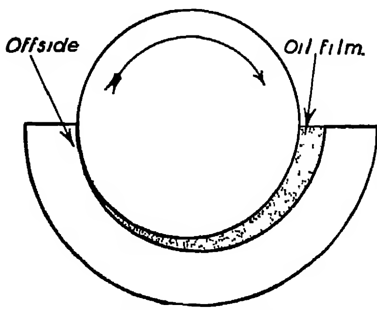


FIG 78A. EXAGGERATED VIEW OF BEARING
Explaining why the greatest wear occurs at one side

Besides machine parts and motor-cars, white met bearings are now used extensively on railway wor and one reason is the ease of replacement when nece sary by remetalling, which we will discuss in th chapter.

The white metal is only a thin layer, and if the bea ing runs hot and melts there is no fear of damag resulting, as the part cannot get away and break othe parts of the machine. It is claimed that die-cast bea ings of solid white metal are very good, and engine have run quite successfully on them, but there

always the risk of the metal spreading, especially at the ends, and as a thicker layer of metal is used, damage to the engine is likely if the bearing runs out. White metalling is not confined to small bearings, as in a bearing from a huge crushing machine 1200 lb. of babbit was required for relining.

CLEANING AND TINNING BEARING SHELL

The bearing should be perfectly clean before the tinning operation is commenced, and also care taken that fumes during the heating process do not foul the surface, therefore always heat the bearing from the back. It cannot be over-emphasized that cleanliness in white metalling is the essential factor. Pure tin is the best solder to use, but in any case only use the very best tinman solder, and killed spirits of salts is the most popular flux. When tinned, dip the bearing in a pail of water, and wipe the surplus metal off with clean cloth. With the water method there is less chance of dirt, fluff, or pieces of cotton getting on the bearing face. For heating large bearings, a muffle should be used, but a gas ring or a blowpipe can be used on small bearings, which comes in handy for heating the jig, and also keeps local parts heated. If ready to proceed, the bearing should again be heated and a thin layer of best solder be given, or if a few pieces are to be done, this can be left until all are ready. This tinning can easily be done by plunging the bearing into the molten metal, but there is always the risk of dirt, etc., contaminating the metal The operation is illustrated in Fig. 79.

TESTING TEMPERATURES

We are now ready for the white metal which, of course, must be above suspicion. If white metal is heated higher than necessary, the metal is useless, and

it should not be repeatedly heated, and then left to cool. Only use sufficient metal to do the job required each day, and if there is any over, pour it out of the melting pot so that it is cooled quickly, and it then can be used on the next work. Two other important points are that the metal should be stirred well, so that

FIG. 79. TINNING A SPLIT BEARING.

the mixture is homogeneous, and the temperature carefully watched. A pyrometer is the correct instrument to use, but often a mechanic cannot obtain one, and must therefore be able to check this without instruments. In place of a pyrometer, small pieces of white paper can be used, which should be dipped into the metal and withdrawn. If the metal is at the correct temperature, the paper will become brown, but not burnt or charred black. Another sign of over-heating is a blue ash, which is an oxide forming on the surface

f the metal. Some engineers cover the surface of the ıetal with a layer of wood charcoal to prevent the xidization. The actual temperature varies with differıt white metals, but all should be heated till they run eely, when pouring should be done.

MOUNTING BEARINGS AND POURING

The next operation consists of mounting the bearing a fixture and pouring the metal. We will first discuss e fixtures. If possible, make the fixture so that the etal can be poured vertically, but in all cases see that ovision is made for extra supply of metal at the top. l the mandrels or pieces in contact with metal should cleaned and smoked to prevent the metal sticking them. Instead of smoking, the part can be dusted th French chalk. The necessary fixture can easily be apted by the mechanic to suit each particular case. ith half bearings, an angle plate can be used, and a lf mandrel, or a piece of sheet metal bent to the shaft ameter instead of the solid mandrel. This is placed ainst the side of the angle plate, and the bearing shell mped up in position. Where extra metal is required the faces, either clay, putty, or millboard pulp can used, or a special metal extension added. The clay also useful to lute up any small inaccuracies at the rious joints. The two photographs illustrate further -ups for casting bearings. In Fig. 82, a bearing is ng cast by means of a half mandrel clamped to a te in the vertical position, and Fig. 80 shows the izontal pouring where a shaft and cardboard end ces are used. In both cases there is plenty of clay ing to prevent leakage of the metal. In certain cases ite metalling is done in position. The bearings on all engines, and lead screw nuts on lathes, are amples. The prepared spindles are mounted in line, l the necessary packings, etc., used, and then the

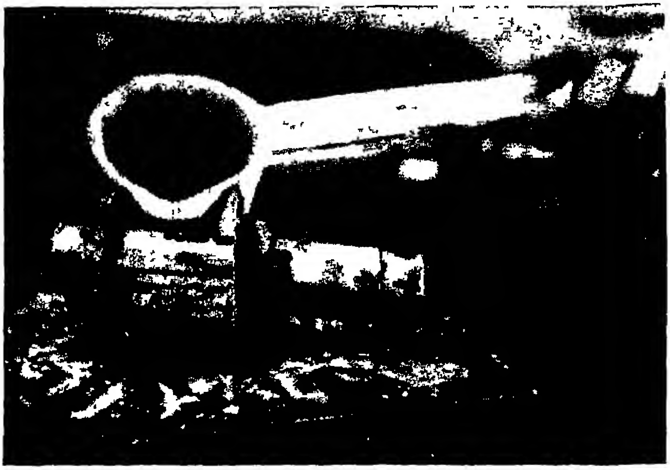


FIG 80. POURING MOLTEN METAL OVER A METAL CORE WITH CARDBOARD END PIECES

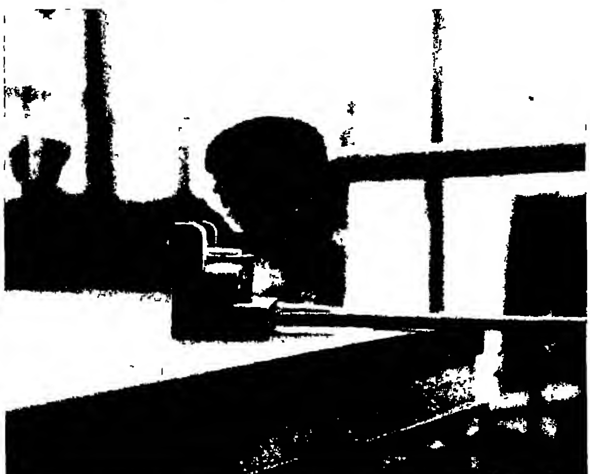


FIG. 81. ANOTHER METHOD OF CASTING A BEARING

metal poured direct. When ready for pouring, the bearing and the fixture should be heated until the tinned surface begins to run, and then hold the ladle as near as possible to the bearing and pour in a steady stream, without any splashing and in one go. Always allow plenty of metal, as surplus can easily be turned off,

FIG. 82. CASTING A BEARING ON A HALF MANDREL

while cavities cannot be filled The bearing should be cooled from the bottom, while the top is kept hot so that the metal can flow down the bearing to fill up any spaces or gas cavities. To help this, dip a clean piece of wire in and out of the part still molten, which will tend to burst the cavities and allow the still molten metal to flow in. This period is only for a minute or two.

In workshops of any importance and where production is large, white metal die-casting machines are used, and the metal is forced into the die by means of a

plunger. All the previously-mentioned points must still be watched, but it cuts out the danger of bad pouring. As shown in the photograph, Fig. 84, the temperature of the molten metal is checked with a pyrometer, so that there is no danger of overheating. The die is fixed to the machine, and swung over an

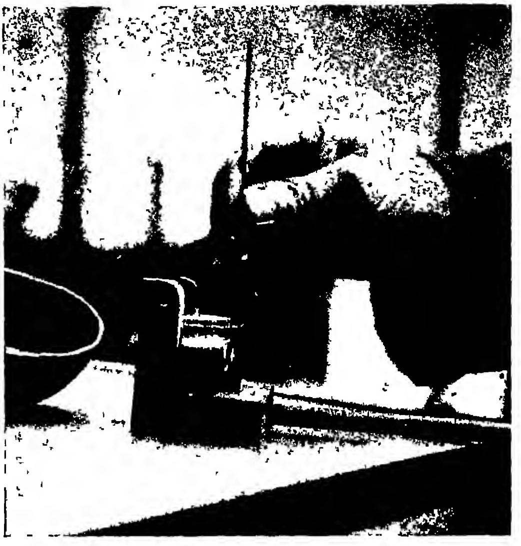


FIG. 83. FREEING GAS CAVITIES

orifice through which the metal is pumped by the hand lever at the side. So that there is no chance of the metal adhering to the die, all the exposed surfaces are dusted with French chalk before each cast is made. The halves of the die are clamped together, and held down to the seating by the hand wheel mounted above the machine. White metal bearings have also been successfully manufactured by centrifugal casting, but at present are not extensively used.

FIG. 84 A WHITE METAL DIE-CASTING MACHINE

TESTING AND MACHINING

A good bearing which has contact with its shell, will ring true when lightly struck with a hammer. When

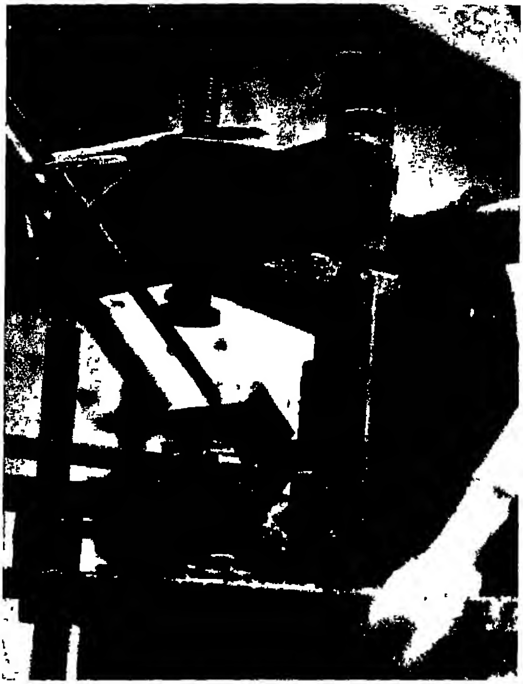


FIG 85. DIE-CASTING MACHINE WITH DIES OPEN

scraping the metal, if the material seems hard and gritty it is a sign of overheating, and the bearing should be scrapped. The processes of cutting oil grooves, rolling, scraping and bedding-in will be considered in another chapter.

DEFECTS AND REMEDIES

If a bearing does not ring true, or the metal has not adhered to its shell, it must be relined, and greater care taken to see that the flame does not reach the tinned surface, and also that the shell is heated enough to run the tinned surface before pouring the metal. Blowholes are often due to the temperature of the metal not being correct, or bad pouring, but can also be due to oil, water, or air being in the bearing, and unable to get away.

OILWAYS AND CLEARANCE

Oilways carry the lubricant from its source of supply to the bearings, and consist of either pipes, channels, or drilled holes, according to the design of the machine. Oilways also collect used oil, and return it to be filtered ready for circulation. The area of the pipe should be sufficient to flood the bearing with oil, and the mechanic must see that all oilways are clear. Many ingenious devices have been made to supply oil to revolving cranks of steam engines, and one which is well used consists of a sight feed held vertical by balance weights. The photograph, Fig. 86, is an underneath view of a petrol engine, and shows the oil channels from which the dippers on the ends of the connecting rods pick up the oil. Used oil drops to the sump, and is then drawn through its filter and pumped around for recirculation. In the swivel pins of motor-car front axles there are usually bushes at the top and bottom, and these are lubricated from one filler. The pin has an oilway running down the centre, which connects to both bushes by small drilled holes. The line drawing illustrates standard practice in auto engine design with regard to oilways, although there are many different methods which act in a similar manner. In the one

illustrated, the oil from the pump is delivered to the main distributing pipe, and then fed to the crankshaft bearings. The crank is drilled, and the oil fed via drilled holes from the bearing to the throws, which lubricates the connecting-rod big end. The various bearings, oil pipe, and oil holes can be seen in the

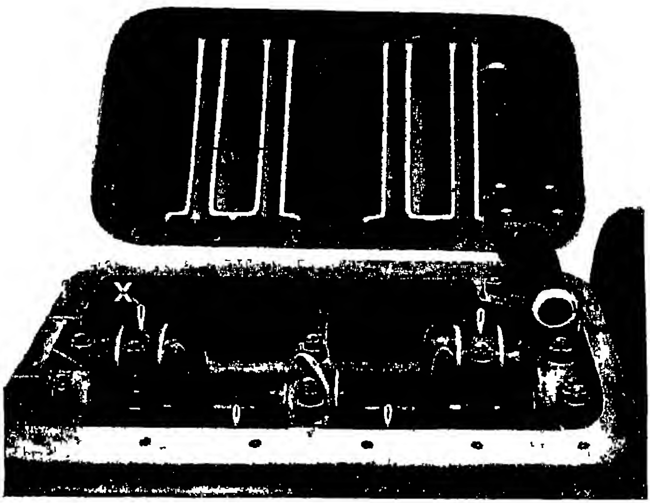


FIG. 86. UNDERNEATH VIEW OF PETROL ENGINE CRANKCASE WITH OIL SUMP REMOVED

photograph, Fig 88, which shows the case with the components removed. So that no leakage of oil will occur at the face when the crank protrudes through the case, an oil thrower is turned on the shaft, which slings the oil back into the case. Even if any escapes past this and tends to ooze out, an oil return groove will again return it to the oil sump. In other engines the oil is fed to the crank pins by pipes, or again by means of channels cast in the crankcase. Bearing clearance is

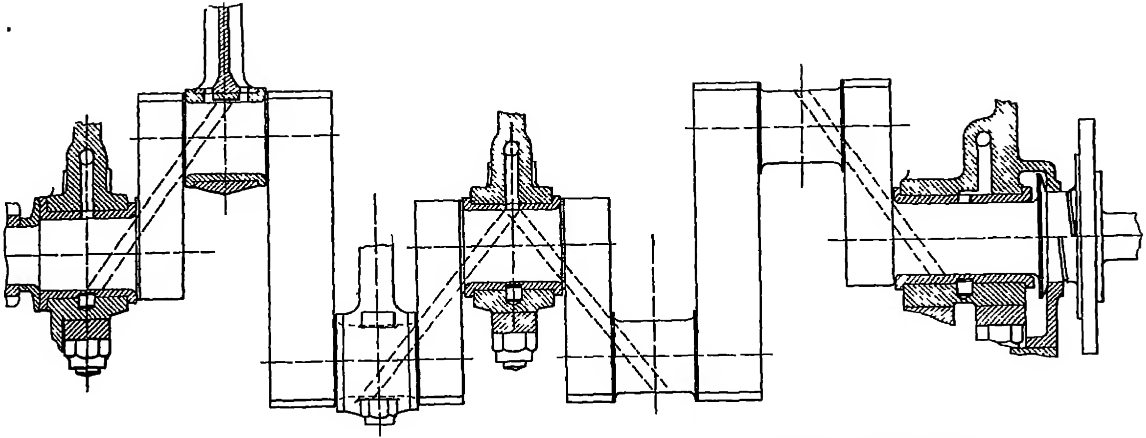


FIG 87. THE BEARING ARRANGEMENT ON A TYPICAL FOUR-THROW CRANKSHAFT

the amount of space which must be left not only to allov
for an oil film, but also for any expansion, etc., whicl
takes place during the running of the engine and machine
No definite figure can be given to suit all cases, an

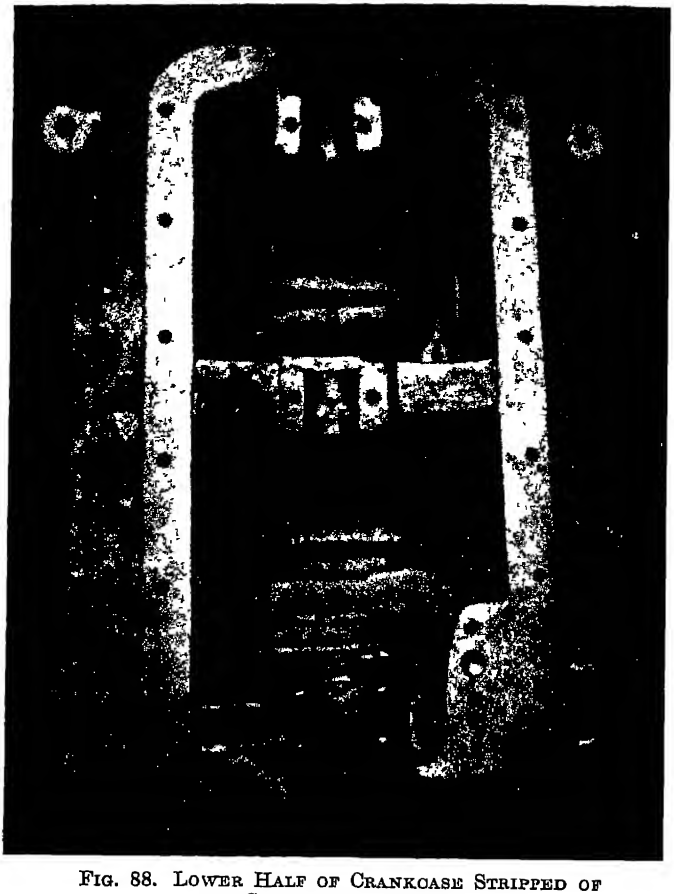


FIG. 88. LOWER HALF OF CRANKCASE STRIPPED OF THE CRANKSHAFT ASSEMBLY

this is often left to the fitter or erector to decide. An average clearance for running fits on medium size shafts is to allow ·001 for each inch in diameter The

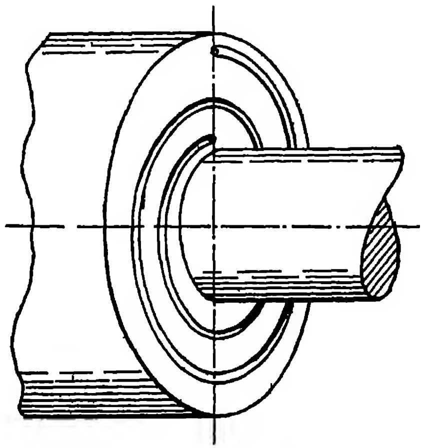


FIG. 89 SPIRAL OIL GROOVES

clearance and limits for various sizes of bearings, so that a good film of oil can be assured, are as follows—

For the Shafts—

Up to 1 in. diameter the size should be	minus 1	to minus	2	thousandths
,, 2 ,, ,, ,, ,, ,, ,,	1¼	,,	2½	,,
,, 3 ,, ,, ,, ,, ,, ,,	1½	,,	3	,,
,, 4 ,, ,, ,, ,, ,, ,,	2	,,	3½	,,
,, 6 ,, ,, ,, ,, ,, ,,	2¾	,,	4¾	,,
,, 10 ,, ,, ,, ,, ,, ,,	3¼	,,	5¾	,,

For the Bearings—

Up to 1 in diameter the hole should be	plus ½	minus	¼	thousandths
,, 2 ,, ,, ,, ,, ,, ,,	¾	,,	¼	,,
,, 3 ,, ,, ,, ,, ,, ,,	1	,,	½	,,
,, 4 ,, ,, ,, ,, ,, ,,	1¼	,,	½	,,
,, 6 ,, ,, ,, ,, ,, ,,	1½	,,	½	,,
,, 10 ,, ,, ,, ,, ,, ,,	1¾	,,	1	,,

OIL GROOVES

The oil grooves distribute the oil over the various faces on the bearings, and a lot of attention is now being paid to these points. The old method was to

FIG. 90. SPIRAL OIL GROOVES ON SHAFT

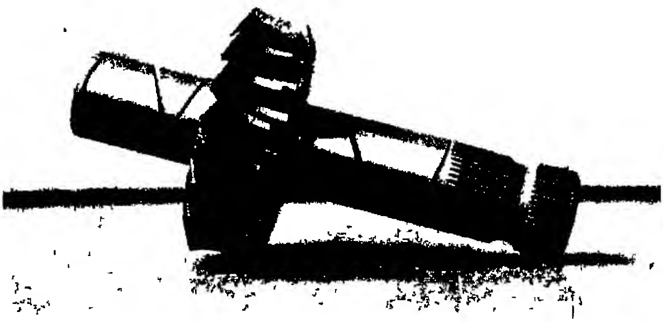


FIG. 91. OIL GROOVE ON WORM WHEEL SHAFT

cut a spiral groove straight round the bearing, without any theory except to distribute the lubricant. The idea of an unbroken film, and also the introduction of Michell bearings with their taper wedge of oil, has

ltered these ideas, and now the oil is led in at least ressure, via an oil groove at right angles to the moving art. Oil grooves should have round edges, should be s few as possible, and must not be allowed to run out

FIG 92 LONGITUDINAL OIL GROOVES ON UNDER-SIDE OF GRINDING MARKING TABLE

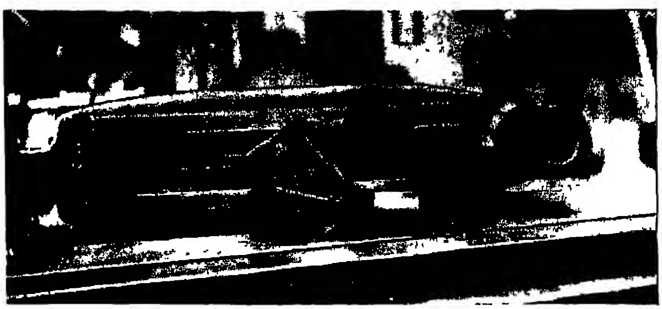


FIG. 93 OIL GROOVES IN THE SLIDES OF A SHAPING MACHINE

the bearing, as otherwise the oil will take the easiest ith, and not do its work. Oil grooves can be put ther on the shaft or in the bushes, but the usual way in the bushes, as these are the softest material. A w types of oil grooves have already been illustrated

under the chapter on motor-car bearings. The groo
if spiral, are cut right and left hand, or sometimes bo
according to the position of the oil inlet (see Fig. !
The faces of thrust bearings are lubricated by eccen
oil grooves which distribute the lubricant over the fa

FIG. 94. CUTTING OIL GROOVE ON GROOVING MACHINE

and others of spiral groove on journals and thrusts a
also illustrated. Oil grooves are also necessary on su
faces as the ways or slides of machines, and the unde
side of a grinding machine table is shown in Fig. !
and the slides of a shaping machine in Fig. 93. Son
times the delivery is at the high pressure side, and it

then necessary to cut a groove round the case, so that the lead in is at point of low pressure. If new bearings are fitted to any machine, first make a note of the

FIG 95 CUTTING OIL GROOVES IN A BEARING BY HAND

oilways, and also reason it out yourself, as oil grooves can be overdone, and also cut the wrong way, with the result that oil is returned and not supplied to the bearing. With white metal bearings, grooves known as mud grooves, consisting of a chamfer along the butt faces of each half bearing, are added. These, of course,

should be of a slow chamfer, so that the oil can be dr
in by a wedge action, and the grooves must not ext
the full width of the bearing, but end from one sixte
to one half, according to the size of the bearing.
grooves in the region of high pressure are not only l
but often harmful, as they break up the film of
The oil grooves can be cut on a lathe with a scr

FIG. 96. OIL-RETURN GROOVES ON END OF AXLE SHAFT

cutting attachment, or in larger works oil groov
machines are fitted which do not require such skil
labour. The fitter, of course, can, if no machines
available, easily cut the grooves by hand, and
operation is illustrated in Fig. 95.

OIL-RETURN GROOVES AND PACKINGS

We have already mentioned and illustrated
return grooves as used on engine parts, and Fig.
shows oil-return grooves on the end of axle shaf
Their use is to stop oil from the differential leaki
along the shaft, and obtaining access to the bra
shoes. These return grooves should not be rounded
with oil grooves, but should be shaped V with t

straight side acting as the return. Oil-return grooves can also be used on vertical bearings to bring the oil from a reservoir up to the bearing. If it is not possible to fit return grooves where shafts project through the cases, other methods, such as felt washers, should be

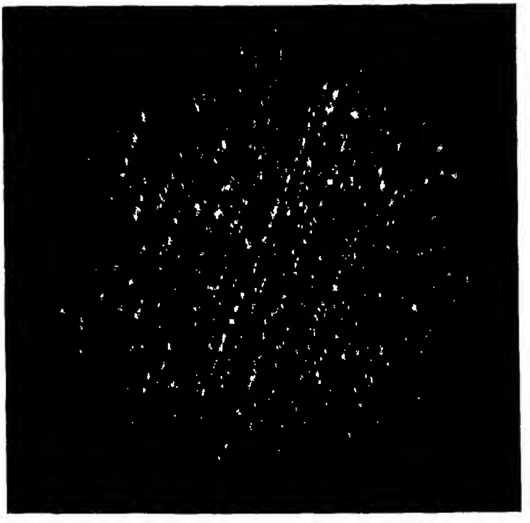


FIG. 97 MICROGRAPH OF GROUND CAST IRON SURFACE

used, not only to save the lubricant, but to keep the machine clean.

On motor engines it is often found necessary to fit baffles at the entrance or bottom of the cylinder bores to reduce the amount of oil splashed upon them.

If more than is necessary for lubrication is delivered, a certain amount will get burnt, or charred, and form carbon deposit, which results in pre-ignition, failure of piston rings, and many other troubles due to contamination of the oil.

FINISHING SURFACES

The surfaces of bearings should be in a perfect condition if friction is to be reduced, and often this means not only special cases, but a lot of hard work. With steel shafts the journals can be ground to size, but

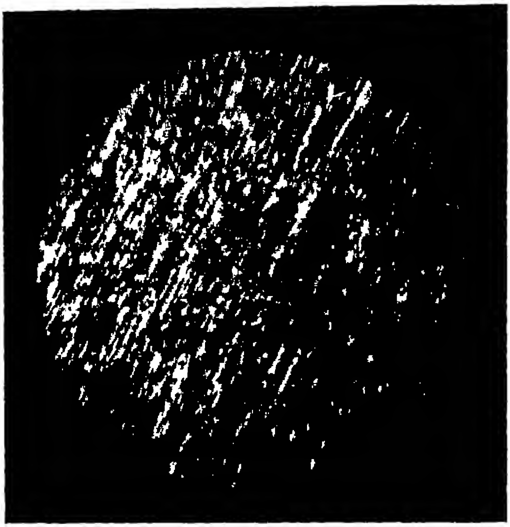


FIG 98 MICROGRAPH OF ROLLED CAST IRON SURFACE

although the surface appears smooth, if examined under the microscope, it appears more like a ploughed field. To give the reader some idea of surface finish, we will take the three photographs in Figs 97, 98, and 99. The first is a surface of cast iron which has been ground, magnified to 25 diameters The other two surfaces have been respectively rolled and polished. The rolled surface has a few of the grinding marks removed, and the structure of the cast iron can just be seen, and in

Fig. 99, which was hand polished, a perfect surface free from scratches is shown. In polishing surfaces with oxides, a chemical action takes place, and material is not removed, but the irregularities are smoothed over. One of the places on motor-car engines where the finish of the bearing surface has improved is the cylinder bore, and the majority of bores are now rolled,

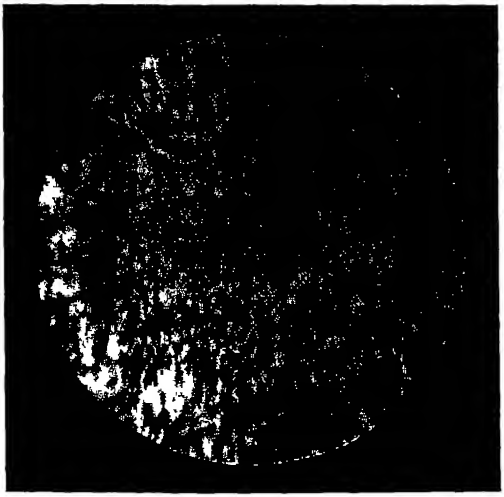


FIG. 99 MICROGRAPH OF POLISHED CAST IRON SURFACE

apped, or lined. A rolling tool is shown in Fig. 100, nd consists of hardened rollers free to revolve on a pindle which is fixed into the bore, and while rotating s reciprocated up and down the bore. It not only produces a good finish, but slightly compresses the metal nd gives it a harder surface. Honing is another nethod, and the hone in Fig. 101 consists of fine stones nounted in a holder, and the action is similar to grinding except that only small amounts of material are

removed. After the surfaces have been attended to. there is the scraping and bedding-in of the bearings, working clearances, and also the alignment of the bearings. A frequent cause of failure in bearings can be attributed to lack of alignment, or a bent shaft which

FIG. 100. ROLLING TOOL

unduly stresses the bearing. If a shaft is weak, or the distance between the supports too great, similar trouble will be experienced. All these points should therefore be watched during fitting and assembly. With connecting rods for petrol engines, this is always checked by the assembler, and fixtures are provided to test the alignment and, if necessary, reset them correctly.

The fitters can be saved a lot of work during the

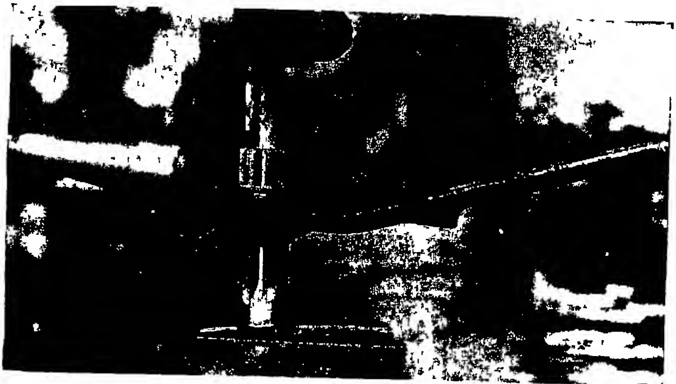


FIG. 101. REAMING THE PIVOT PIN BEARINGS ON THE STUB AXLE OF A CAR

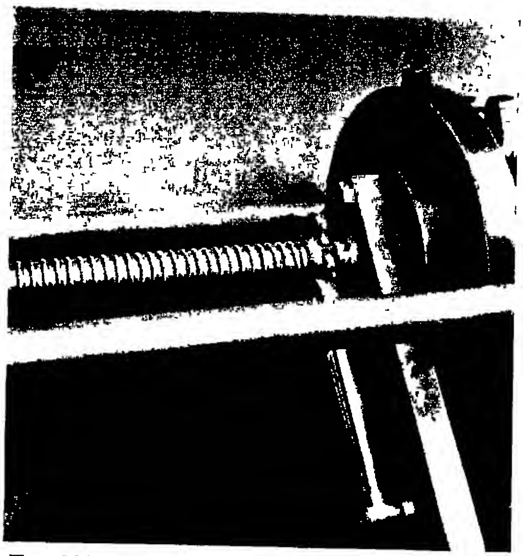


FIG. 102. BROACHING CONNECTING-ROD BEARING

assembly if all bearings are reamed in line and, if possible, broached as illustrated. The type of broach with rounded burnishing teeth can also be used in

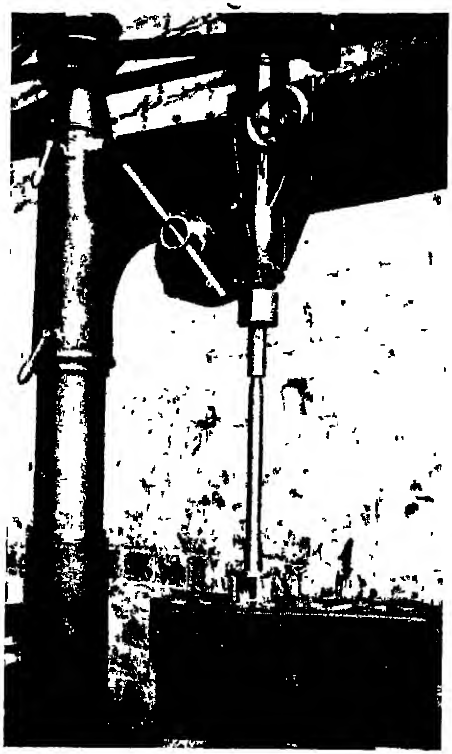


FIG 103 HONING TOOL

many other parts for finishing round holes, such as valve guides, tappet guides, etc. Fig. 101 illustrates the two phosphor bronze bearings on the stub axle of a motor-car being reamed in line after assembly. Often the working clearance is given by means of packing

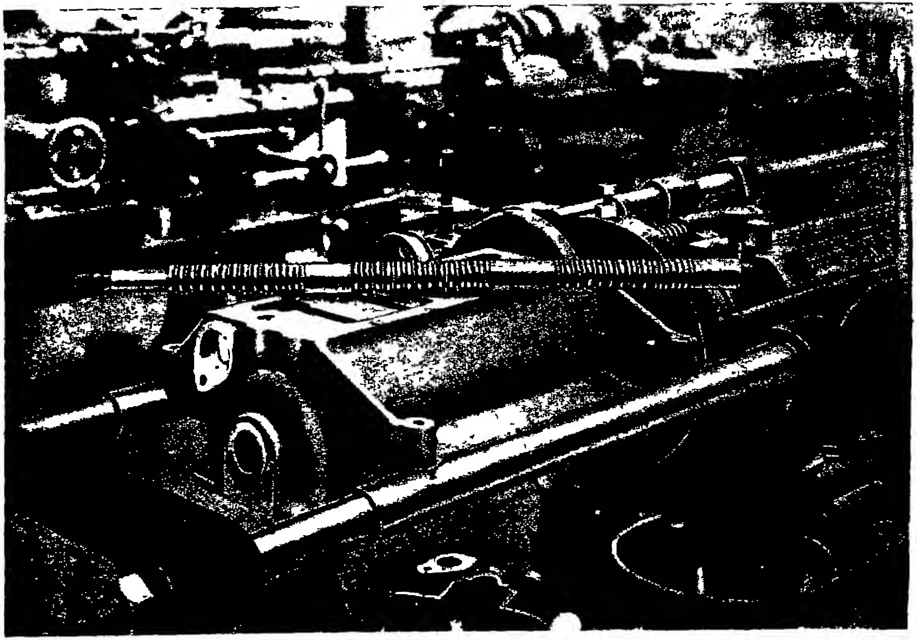


FIG 104. BROACHING A THREE-BEARING CRANKCASE

shims, and this is the system used in the majority of connecting rods, and an illustration has been given in previous chapters under Motor-car Bearings The bores of white-metalled connecting rods, and in a few cases even the main bearings of crankcases, are broached to size before bedding-in, and when dealing with white metals, the last teeth of the broacher are rounded and

FIG. 105. GROUP OF BEARING SCRAPERS

act as burnishers or rollers, and compress and also harden the surface

SCRAPING AND BEDDING-IN BEARINGS

When the shaft is true, correct to diameter, and has a good finish, the operation of bedding-in can be commenced. An an illustration, we will take a three-bearing crankshaft for a petrol engine. First, the tools must be described. They consist of hardened steel with cutting edges, and their shape depends upon the articles being scraped. Flat ones similar to chisels are used for flat surfaces, and round ones for journal bearings, but the reader will gather more information by referring

,o Fig. 105, which illustrates a collection. They are ›asily made out of old files, and the edges should be kept .harp. Even when in use, an oil stone should be kept

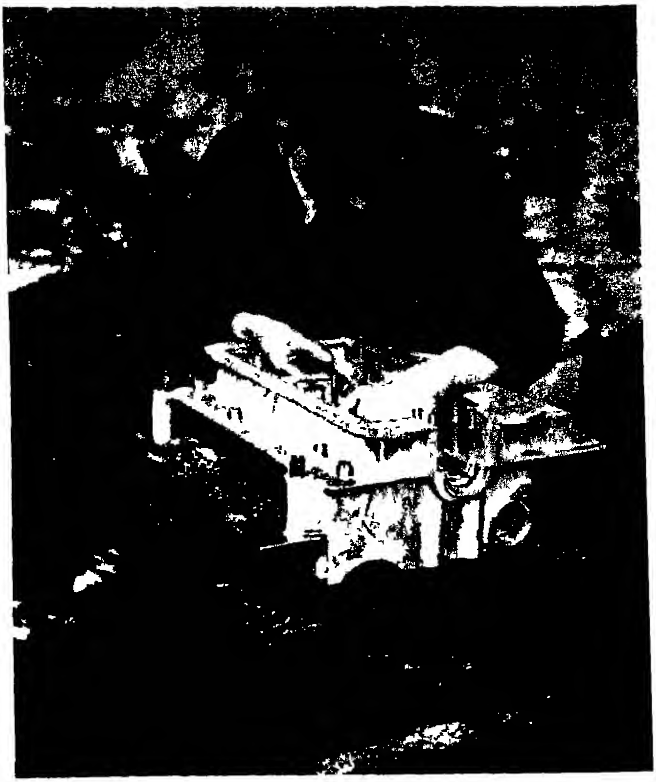


FIG. 106. SCRAPING THE CRANKSHAFT BEARING IN POSITION

ındy so that a few light rubs can be given. The half əarings in the case are first put in position, the cap əaring in all cases being left till later The crank or

FIG. 107 BEARING JOURNALS BEFORE (LEFT) AND AFTER (RIGHT) SCRAPING

FIG 108 TOP OF CONNECTING ROD BEARINGS

Fig. 109 Typical Factory Plant for Running-in Petrol Engines

standard bar is then lightly smeared with a marking paste, such as Prussian blue or red lead, and then laid in the bearing and revolved while being held by pressure in the bearings. When lifted out of the bearings, they are marked on the high places The various scrapers are now taken, and the high points reduced by removing very thin shavings of metal. The process of scraping is more difficult than the description sounds, and experience is necessary before good work is produced. The operation is illustrated in Fig. 107, where two journals are shown, of which the left hand is marked ready for scraping, and the right hand is after scraping. The marking of the bearings and scraping is continued until the marks are distributed over the whole bearings. When, as in the example, there are three bearings, much more care is needed, and the centre bearing should not be placed in position till the two end ones have been finished The next procedure is to fit the caps, and here two operations are necessary: first, the bearing surface needs scraping, but at the same time the faces at the junction of the two halves need levelling and also reducing, so that the cap is in contact with the shaft. The tightness of the shaft should not depend upon the tightness of the nuts, but on the clearance allowed when fitting With white metal bearings the shafts should not be free enough to be spun round, but with phosphor bronze bearings, which will not give like white metal, the shaft should have no play, but be free enough to spin round. Fig. 108 illustrates a connecting rod and bearing in which the fit depends not on packing shims, but the fitting of the two halves until the correct clearance is obtained.

RUNNING-IN BEARINGS

White metal bearings should be run-in for a short period, and then the bearings can be checked again

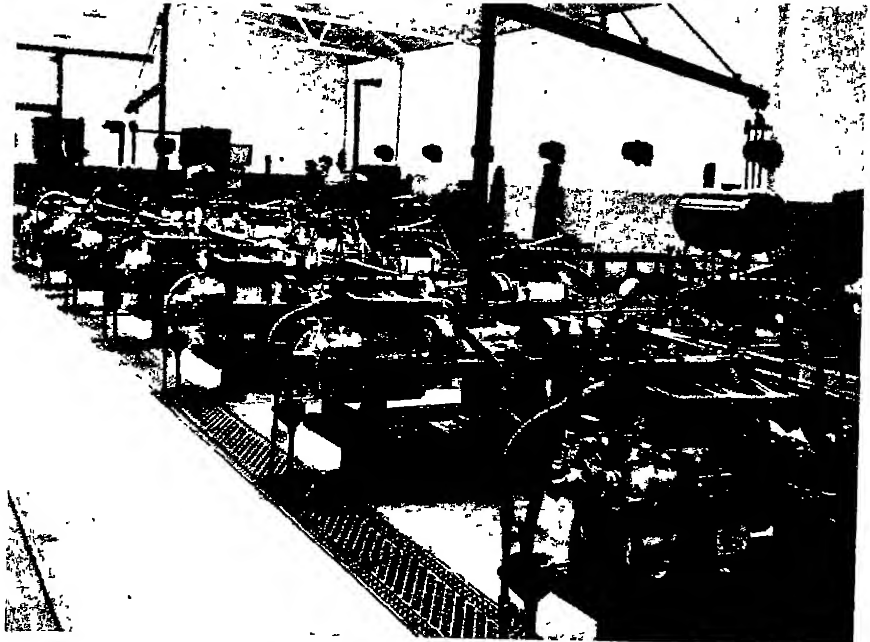


Fig 110 Another View of a Batch of Engines Being Run-in

This running-in allows the white metal to form a perfect bearing, the small cubic crystals of hard material sinking into the softer mass, and although the bearings were slightly stiff before, they should be free after running-in. This, of course, must not be done with bronze bearings as the material is too hard, and the bearings must be free to start with. The two illustrations, Figs. 109 and 110, show a plant for running-in motor-car engines after scraping and fitting. As these were of high standard, every bearing was again examined after this operation to make sure that surfaces were perfect, and all had obtained the correct lubrication When running-in see that a plentiful supply of lubricant is available, and control the heat of the bearings also, which can easily be managed by a stream of oil flowing over them In replacing bearings, watch the oil holes, and see that they register with the supply, and, if remetalled, see that no metal has got in the pipe or choked the supply in any way After fitting bearings, see that they are protected from all dirt and dust which causes the wearing of the surfaces, and, lastly, remember that a bearing must be lubricated , also if this is not automatic, methodical oiling must be carried out with correct lubricant at regular periods

SECTION XXXI

BALL AND ROLLER BEARINGS

BY

A W JUDGE, WH SC, A.R.C S, A M.I.A E.

SECTION XXXI

BALL AND ROLLER BEARINGS

INTRODUCTORY

ALTHOUGH the plain type of journal and thrust bearing has been developed to a fair standard of efficiency by careful attention to such factors as the choice of suitable bearing metals, the proper bearing load pressures and the correct means of lubrication, the fact remains that the plain bearing depends upon sliding resistance of two solids separated by an oil film. As tests have shown conclusively, the efficiency of such bearings has reached its limiting value in the case of whitemetal bearings and hard steel shafts, or journals, running in oil fed under high pressure through suitable oilways In the search for still further improvement in bearings, more particularly in the case of motor-car and cycle ones, it was discovered that by substituting rolling friction for sliding friction in bearings a considerable reduction in friction could be obtained. To appreciate the principle underlying the action of a ball or roller bearing, one may consider a heavy metal object having a flat lower surface resting on a flat metal table—the "marking-off" table of the engineer's shop, for example. If one attempts to drag this object across the table a certain appreciable effort will be required This effort will be less if the table top is greased or oiled. If, however, one places metal rollers under the metal object, as shown in Fig. 1, the force required to move it across the table will be found to be appreciably less. The approximate forces required to move the object under the above conditions are indicated in the diagrams (Fig. 1). By

lubricating the rollers and the top of the table a still smaller effort will be found to be required.

BALLS AND ROLLERS

If the rollers are replaced by hardened steel balls a somewhat similar saving in effort is obtained, but if the object to be moved is relatively very heavy, unless a large number of balls is provided, the balls being in

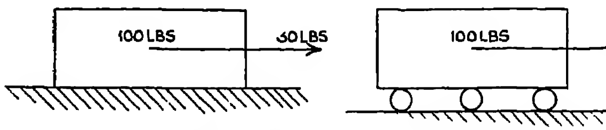


FIG. 1. ILLUSTRATING (LEFT) SLIDING, AND (RIGHT) ROLLING FRICTION

contact only at points may indent the surfaces or themselves be distorted

We thus see that for heavy loads rollers are better, whilst for light loads balls may be used satisfactorily. In the same way one always uses either double row ball bearings or roller bearings for heavy journal and thrust bearings in engineering practice.

THE ADVANTAGES OF BALL BEARINGS

The engineer having to handle, fit, adjust, and maintain in service ball and roller bearings should certainly be acquainted with the more important properties and advantages of these bearings We have already referred to the reduced friction of ball and roller bearings compared with plain ones, and in this respect cannot do better than quote from some interesting tests made on Hoffmann ball bearings as used for line shafting hangers.

A shaft 2½ in. diameter was supported at each end in plain gunmetal bearings, one set being fitted with needle lubricators, and a second shaft with two oil

rings for automatic lubrication, another $2\frac{1}{2}$ in. shaft had Hoffmann ball bearings. A total load of 2,300 lb. was supported in each case. The tests were carried out for several hours so as to obtain steady conditions. Measurements were made of the friction in each case, and it was found that at 250 rev. per min, the two sets of plain bearings gave frictional values which we will represent by the numbers 160 and 130 respectively, whereas the ball bearings gave only 15.* These results prove definitely that the friction of the ball bearing was only about one-tenth that of the plain bearings. A similar statement applies also to the low frictional losses experienced in roller bearings.

The amount of power wasted in line shaft transmission where plain bearings are used is considerable. Although some authorities put this loss as high as 50 per cent of the power transmitted, we are inclined to be a little more conservative and accept the loss figure as 25 per cent, i.e. one-quarter of the power transmitted in line shaft bearings By substituting ball bearings for the plain ones, this loss of power, as we have shown by the test results quoted, is reduced to one-tenth of its value It will thus be seen that only one-fortieth of the power transmitted, instead of one-quarter, is lost by friction at the bearings . this represents a big saving in power.

Quite apart from their greatly reduced friction, and therefore saving in power, ball bearings are much *more compact*, occupying only a small width, although of greater overall diameter than plain bearings.

They are *more reliable*, for if protected against dust and dirt, and properly lubricated once in a while, they are immune from breakdown. *They are cleaner*, for the lubricant consists of a special grease that does not run

* The coefficients of friction in the three cases were, respectively, ·016, ·013, and ·0015

out, and, unlike plain bearings, there is no constant flow and drip of oil over adjacent parts. In industrial factories—more particularly in those engaged in the preparation of foodstuffs, chemicals, and in textile work, this enhanced cleanliness is a big advantage

As regards *lubrication*, there is a considerable saving both of lubricant and of time. It is only necessary, in the case of line shafting and many machines fitted with ball bearings, to pack the bearings with grease when they are first fitted, and then to inject grease—using a grease gun—once every six months. The saving effected in this respect over the cost of daily lubrication work and oil for plain bearings is about 75 per cent, according to figures furnished by Skefko bearing users

Another advantage with ball bearings is that *much higher speeds* can be obtained in many cases, for example, with line shafting, so that maximum production and economy can be effected ; there is also a greatly reduced risk of bearings running hot and seizing up.

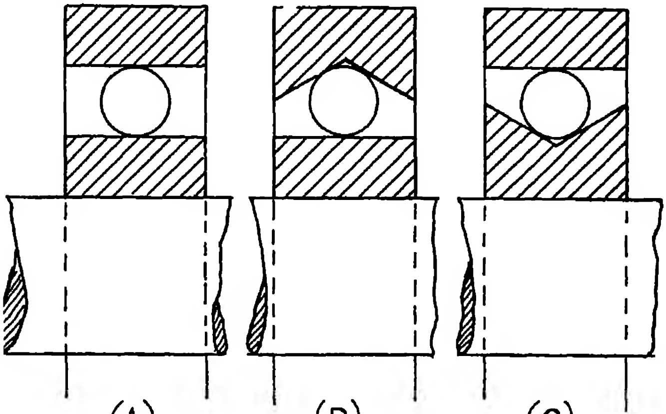
BALL BEARING LOADS AND SIZES

In what follows we shall not attempt to deal with the theory of ball and roller bearings, or to discuss the results of a large number of investigations that have been made by various authorities, but shall give just sufficient information on the subject to enable the reader to ascertain the best type of bearing to use for each particular job, and to find out the most suitable size, method of fitting, adjustment, and maintenance.

The earliest form of ball bearing was that illustrated in Fig 2 *A*, and consisted of a pair of annular cylinders made of hardened steel with a row of balls between , as many balls as possible were used. This "two-point" bearing is the simplest type of a rolling friction bearing, but it has the drawback that there is nothing to prevent

the balls from moving endwise , this bearing, therefore, will take no end load

In order to overcome this drawback, a " three-point " bearing of the type shown in Fig 2 *B* and *C*, was next devised, so that the two points of contact of the ball on the grooved member were on the same radius from the



(A) (B) (C)

Fig 2 Early Forms of Ball Bearings

shaft centre With this type there was a tendency for the grooves to flatten at the places of contact, so that instead of there being point contact—as theory requires —there was surface contact, and consequently a sliding action. After various other possible forms of ball bearing, including " four-point " contact and grooved races (having grooves equal in curvature to the curvature of the balls themselves), the final form of journal ball bearing adopted was that illustrated in Fig. 3. In this case the ball races are grooved to a larger radius than that of the balls themselves. Prof. Stribeck, one of the earliest authorities on ball bearings, who made a large number of experiments on behalf of the German

Small Arms Factory, recommended that the radius of each of the grooves of the ball races should be from $1\frac{1}{8}$ to $1\frac{3}{8}$ times the radius of the balls

There is another important matter in regard to the rubbing of the balls themselves, for if they are packed into the space between the races so as just to touch,

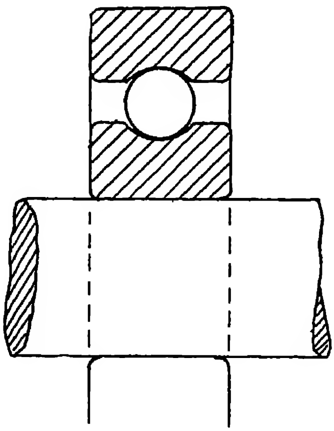


FIG. 3 THE MODERN BALL BEARING

then when the inner race, say, is rotating, the balls will roll around the outer race, and as each ball will be moving in the opposite direction to its neighbour at the point of contact with the latter, there will be undue sliding or rubbing between the balls, causing them to wear more quickly. In order to overcome this difficulty the balls are separated from each other by means of a separate member termed the "cage" This is usually a skeleton structure of soft steel or bronze, having apertures or spaces for the balls (Figs. 4 and 5). Other examples of cages will be found in the illustrations of different commercial makes of ball, roller, and thrust bearing reproduced hereafter

In regard to the question of suitable sizes of ball earing to carry definite loads, although one can calulate the size of ball to suit a given load, there is the uestion of speed of rotation and fatigue of the material

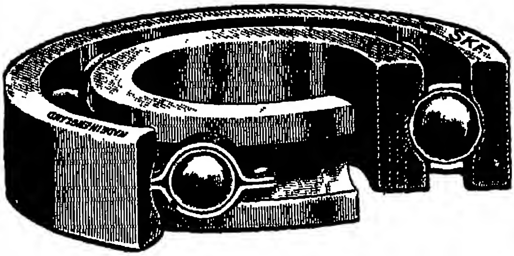


FIG 4 THE SKEFKO DEEP GROOVE BALL BEARING, SHOWING THE BALL CAGE

t is always advisable, therefore, to accept the manuacturer's recommendations for the correct sizes and ypes of bearing to suit specified conditions. Each ball earing manufacturer issues, in tabular form, very

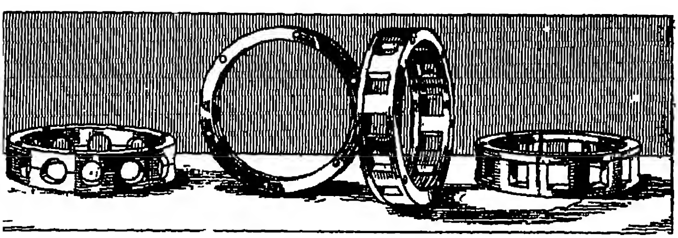


FIG 5 SHOWING CAGES FOR BALL AND ROLLER BEARINGS

omplete information of ball bearing sizes (together vith the actual dimensions of the ball bearings, for the esigner's use) and the safe working loads for these.

In this respect it may be added that the safe load epends primarily upon the diameter of the ball used in

the bearing, and that the load carried increases with the square of the diameter. Thus, according to Prof Stribeck's values, a $\frac{1}{8}$ in hard steel ball has a safe load of 43 lb , a $\frac{1}{4}$ in. ball 175 lb., a $\frac{1}{2}$ in. one of 700 lb., a $\frac{3}{4}$ in. one 1585 lb., and a 1 in. ball 2830 lb.

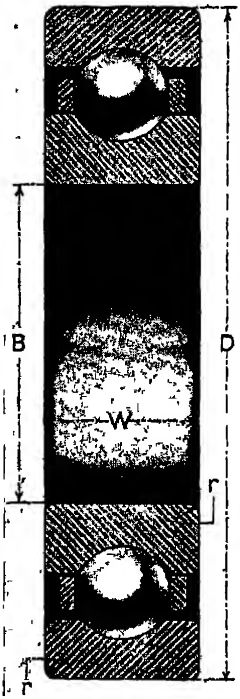


FIG. 6. THE HOFFMANN SINGLE ROW JOURNAL BEARING

TYPES OF JOURNAL BALL BEARING

There is a number of different types of journal ball bearing used in engineering work, each type having its own particular application. The simplest of the commercial types is the single row journal ball bearing, as shown in Fig. 6 These are manufactured in a wide range of sizes and in three distinct types, known as the light, medium, and heavy types. In the case of the Hoffmann standard bearings, the sizes in the light type range from those suitable for fitting over $\frac{1}{2}$ in. diameter shafts up to 10 in. diameter shafts, the corresponding safe working loads ranging, respectively, from 180 lb. to 2320 lb. at 1000 rev. per min. for the $3\frac{1}{4}$ in diameter size. At lower speeds rather greater, and at higher speeds rather smaller loads are carried.

In the medium type, the shaft sizes range from $\frac{1}{4}$ in. diameter up to 6 in. diameter, the corresponding loads at 1000 rev. per min. being 260 lb. up to 2250 lb. for the

2 in. diameter size. The larger sizes of bearing carry proportionately greater loads, but the exact values depend upon the nature of the work and method of mounting.

In the heavy type, the shaft sizes range from 17 mm (0·67 in.) up to 100 mm. (3·94 in), the safe loads varying from 720 lb. at 1000 rev per min. in the former case up to 2090 lb. for the 40 mm (1·58 in.) size The same remarks apply to the larger sizes of bearing as to the medium type.

Other types of journal ball bearing of the single row type available are the extra light type ($1\frac{1}{2}$ in. to 12 in. shaft diameter), the magneto type (5 mm. to 19 mm. shaft diameter in fifteen different sizes), small journal bearings ($\frac{1}{4}$ in. to $1\frac{1}{4}$ in. shaft diameter in nine different sizes) and motor-cycle ball bearings.

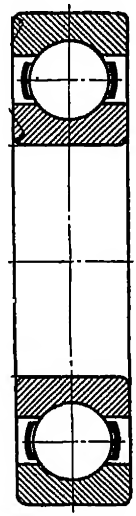


Fig. 7.
The Skefko Deep Groove Single Row Ball Bearing

The Skefko single row, deep groove bearings (Fig. 7), are made in the narrow, medium, heavy, and standard types, and cover a similar wide range of sizes to those previously given. This type of bearing is characterized by deep uninterrupted raceways, which enable it to carry thrust loads in either direction in addition to radial loads. There is no ball filling slot in this type of bearing, so that either face can be arranged to face the load, the bearing thus being fool-proof in assembly, and reversible in operation

THE MOUNTING OF BALL BEARINGS

From all points of view it is essential that ball bearings be fitted to the parts in, or on, which they have to run, in the correct manner In the past the

instances of ball bearing failure investigated have almost always shown that the cause of the trouble has been due to bad fitting and incorrect mounting. Since the successful running of a ball bearing depends upon

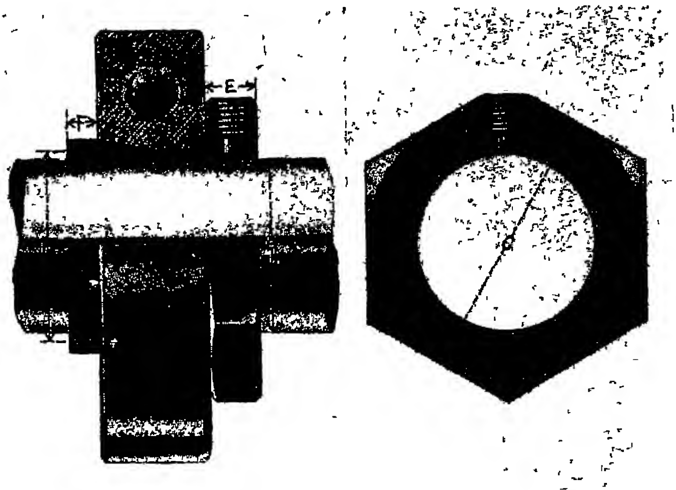


FIG. 8 THE HOFFMANN TAPERED HOUSING FOR BALL BEARINGS

the manner in which it is mounted, we shall give rather more attention to this subject.

ACCURATE MACHINING ESSENTIAL

In the first place, too much emphasis cannot be given to the matter of accurate machining of the parts for taking the ball bearings, namely, the shaft and the housing. The latter items should be turned or ground perfectly cylindrical, with their shoulders, flanges, or abutments quite smooth and square with the shaft. All recesses, flanges, and collars should be concentric with the axis of the shaft. Split housings, collars, or

adapters should be avoided, for these, if badly fitted, are apt to cause distortion of the races.

It is possible, although not always desirable, to employ split housings or bushes when fitting ball bearings under difficult conditions Where such conditions demand a split housing the cap should be registered or dowelled, and the recess for the bearing machined with the cap bolted in position

ALUMINIUM HOUSINGS

If the bearings are to be fitted in an *aluminium housing* it is desirable, where heavy loads are concerned, that the outer race should be housed in a liner of some harder metal The tendency with aluminium housings is for the relatively soft aluminium to be hammered or compressed, thus causing the hard outer race to creep and finally to rotate in the aluminium housing, the play thus developed may rapidly damage the rotating parts, more particularly as the alignment is altered If the shaft is connected to a gear wheel, the teeth of the latter will then not mesh properly with those of the driven gear We have experienced instances of this particular trouble in the case of the bevel-drive shaft bearing housing in the aluminium casing of a light car back axle. Fracture of the teeth of the bevel pinion and crown wheel occurred, and on examination it was found that the outer races of the bevel pinion had actually been rotating in their housings in the aluminium casing.

THE CREEP OF BEARINGS

The creep of the "fixed" member of a ball bearing that sometimes occurs is not due to the friction of the bearing as a whole, but to bad fitting, machining, or unsatisfactory material of the housing

The conditions to be fulfilled by a journal bearing are

such that *the load can never be evenly distributed around the race*. In normal examples of journal bearings the load is vertically downwards, and is taken on one portion (of about 120°) of the outer fixed race, whilst on the inner revolving race this point of maximum load will constantly be changing as the shaft rotates. It is therefore the rotating race of a journal bearing that tends to creep. When the outer race material is not soft there is no tendency for it to creep ; indeed, in many cases the outer race is merely a push fit in its housing and free to move sideways by a limited amount.

VIBRATION AND BALL BEARINGS

Where creeping becomes pronounced in a stationary journal race it is an indication that the revolving mass is out of balance. This means that the point of greatest load travels round the stationary race, pressing it against the bore of the housing and causing creep. The effect on the bearings of a machine resulting from want of balance is not always fully appreciated. To secure steady running with a minimum of vibration, high-speed machines should be properly balanced, both when stationary and when running at normal speeds, i e. both statically and dynamically.

The best method of preventing creep is to make the revolving race of a journal bearing an "interference" fit (or tight press fit) on its seat Given reasonable care in the machining of the shafts and fitting of the bearings, there should be no difficulty whatever in this, and creep can be prevented with the heaviest loads. Keys and similar devices are quite unsatisfactory in this respect, and quickly wear away under the constant chafing

PROPER FIT ALLOWANCES FOR BALL BEARINGS

Before discussing the subject of the practical mounting of ball and roller journal bearings it will be necessary

to consider the types of fit, i.e. the machining allowances that must be employed to give the best working conditions. The usual type of ball bearing consists of an inner and outer hardened steel race, the balls and their cage. In all journal type bearings the fixed or *stationary ball race is made a sliding fit* on its shaft or in its housing. On the other hand, the *rotating member of the bearing must be made a tight press fit* (*or interference fit*) on its shaft or in its housing, as the case may be, in order to prevent creep. Each ball bearing manufacturer issues instructions or tables giving the allowances of the shafts, bearings, and housings for their particular types The following information, although applicable to most types of journal bearing, is based upon the recommended procedure of Messrs. Hoffmann, Ltd

In the case of the interference fits of *light type* journal bearings up to 4 in., and *medium* and *heavy type* bearings up to 2 in. shaft diameter, should the race be on the high limit and the shaft on the low limit the dimensions will be exactly the same in both cases, so that the fit will be too easy. On the other hand, when the bearing is on the low limit and the shaft on the high limit there will be a difference of ·001 in , which will give too great a fit allowance for these bearings In actual practice, however, it will be found that these extreme cases rarely occur. The majority of bearings are on the low side of standard in the bore and, as the shafts are likely to be well within the limits, in most cases the fit will be correct. In other cases, another bearing can be selected, or if the shaft is on the high limit it can be reduced slightly.

TABULAR INFORMATION ON BEARING FITS

The following tables will be found very useful in connection with machining shafts and housings. If the dimensions or allowances given are adopted by those

who have to machine parts for taking ball bearings, the most satisfactory results will be obtained

In connection with these tables it will be observed that Table I refers to the *inner races* of *both* ball and

TABLE I

INNER RACES OF BALL AND ROLLER BEARINGS

Type of Bearing	Bore of Inner Race (Inch and Metric Sizes)	Diameter of Shaft	
		Interference Fit (Rotating Shaft)	Sliding Fit (Stationary Shaft)
Light	Up to and including 4" or 100 mm	+ 0002" to + 0007"	+ 0000" to − 0005"
Medium or heavy	Up to and including 2" or 50 mm		
Light	Over 4" or 100 mm	+ 0005" to + 001"	− 0002" to − 0007"
Medium or heavy	Over 2" or 50 mm and up to 4" or 100 mm		
Medium or heavy	Over 4" or 100 mm	+ 0007" to + 0012"	− 0005" to − 001"

TABLE II

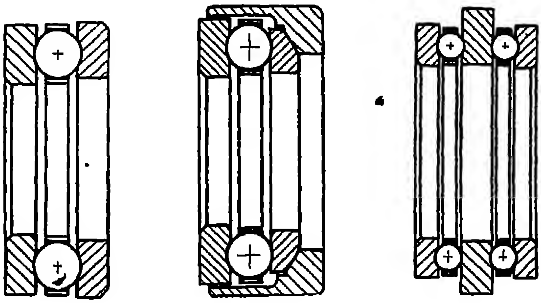
OUTER RACES BALL BEARINGS (JOURNAL TYPE)

Type of Bearing	Outside Diameter of Bearing	Bore of Housing	
		Interference Fit (Outer Rotating Race)	Sliding Fit (Stationary Outer Race)
Light or medium	Under 2"	− 0005" to − 001"	+ 0000" to − 0005"
,, ,,	2" to 3"	− 0007" to − 0012"	− 0002" to − 0007"
,, ,,	3" to 5"	− 001" to − 0015"	− 0005" to − 001"
,, ,,	5" and over	− 0015" to − 002"	− 0005" to − 001"

roller journal bearings (Hoffmann), and Table II to the outer races.

THE FIT OF THRUST BEARINGS

In the case of thrust bearings (Fig. 9), as the load on the balls is in the direction of the shaft axis, there is no load present to search for any clearance between the



A B C

FIG 9 EXAMPLES OF THRUST BALL BEARINGS

A = Single thrust type
B = Aligning single thrust type with outer housing
C = Double thrust type

ball race and the part that it fits. It is therefore not necessary to make the races of ball thrust bearings a tight fit either on, or in, the revolving or the stationary member As a general rule the revolving race is a push fit on, or in, the revolving member, whilst the fixed race is a push fit in its housing

It is essential that the stationary housing be truly concentric with the rotating member's axis.

Further (Fig. 10), it is necessary to provide flanges, collars, or abutments for transmitting the thrust from the revolving member to its revolving thrust race, and from the other thrust race to its stationary housing

Unlike journal type ball bearings, ball thrust bearings

do not give rise to creep, as push fits are employed, but cases sometimes occur of thrust races creeping up against their abutments. If the stationary abutment is out of square with the axis of rotation, most of the load is transmitted to the revolving race at the place where the two races are closest together. This causes the point of maximum load on the revolving race to

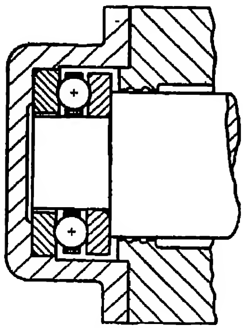


Fig 10
Showing Method of Mounting Single Thrust Bearing

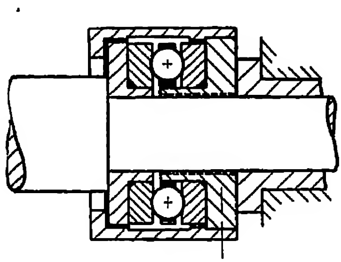


Fig 11.
A Single Thrust Bearing Mounting for Plain Journal Bearings

revolve as the race revolves. This, in turn, produces a revolving point of maximum load between the revolving race and its abutment, and "creep" then occurs One should therefore look for the cause of "creep" in the non-alignment of the abutment of the opposite race. Another important point in connection with the fitting of thrust ball bearings is that the rotating member—the shaft, for example—should run perfectly true in its journal bearings. With plain bearings this is not always an easy matter to ensure, so that wherever possible journal ball or roller bearings should be used.

When it is not possible to employ journal ball or roller bearings the thrust bearing must be protected

ɔm the effects of wear in the plain bearings, as this ɘar will cause inaccuracy of alignment Fig. 11 shows typical arrangement of a good method of protection this case both races of the thrust bearing are centred ɔm the revolving shaft. If the plain journal bearing ɘars and allows the shaft to drop, both of the thrust ces drop with the shaft and the tracks still remain

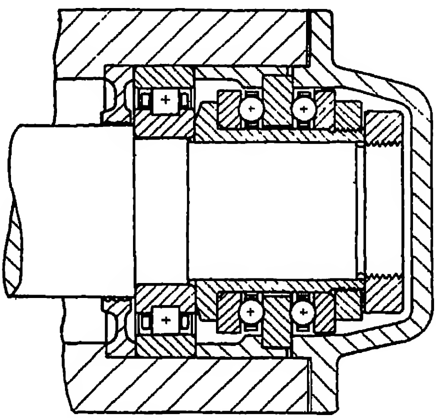


FIG 12 A TYPICAL DOUBLE THRUST BEARING MOUNTING

ncentric. It should be pointed out that the sleeve on ıich the stationary thrust race is mounted acts as small journal bearing, taking the weight of the ıtionary race.

THE FIT OF DOUBLE THRUST BEARINGS

As we have already observed, the central ring of the linary double thrust bearing (Fig. 9 *C*) is thicker, d of larger diameter than the other two races. This ıtral ring is usually clamped tightly to the fixed using by means of a split cap or clamping ring

provided with a recess to take the central ring near its outer periphery

The other two outer races must rotate with the shaft, and at the same time be quite free from end play. These conditions necessitate the use either of a clamping nut and a shoulder on the rotating shaft or, what is much better practice, as there is less risk of clamping the balls

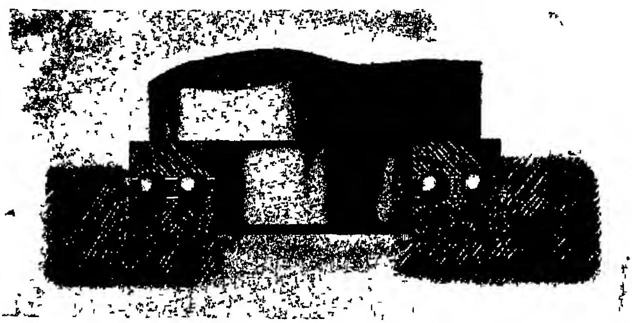


FIG 13 THE HOFFMANN DOUBLE ROW, HEAVY DUTY THRUST BEARING

too tightly against the races, of a separate screwed sleeve with its own shoulder or flange and nut. Such a unit can be supplied, properly adjusted and locked in position, ready for slipping over a shaft and clamping up against a suitable shoulder on the latter by means of a nut screwed on to the threaded part of the shaft (Fig. 12). The clamping pressure in this case is not transmitted through the balls and their races, but only through the material of the sleeve. This is no doubt the better method, as it relies upon the ball bearing manufacturer adjusting the thrust races to the correct clearances, and also as it relieves the thrust bearing of only clamping stresses.

In the example shown in Fig. 12 the sleeve carrying the double thrust bearing abuts against one face of a

journal roller bearing, taking the journal load at this end of the rotating shaft. The complete thrust unit (with its sleeve) is held against the face of the inner ring of the journal bearing by means of the clamping nut shown on the left of the shaft. The central ring member of the thrust race is clamped securely and centrally to the stationary, or outer, housing by means of the cap shown. The latter has a register boss or ring to enable it to enter the right-hand portion of the housing quite centrally. This "locating" ring method of ensuring centrality of a fixing member is widely used in ball bearing design. It should be noticed that the clamping cap does not bed flush with its fixed member on the right, but that a small "clamping" clearance is left.

DOUBLE ROW, HEAVY DUTY THRUST BEARINGS

Where particularly heavy end loads or thrusts come upon rotating shafts, as in the case of pedestals, turntables, and crane posts, it is usual to employ a special type of thrust race having two rows of balls, as shown in Fig. 13 In this example the moving race has two concentric grooves for the balls, whilst the fixed or lower race is made in two separate parts that rest upon a ring having annular faces on the top side and flat at the bottom This ring is purposely made elastic by means of a large number of saw cuts made radially in opposite directions and extending almost entirely through the section. The thrust from each row of balls is carried directly on this ring, and the load is thus distributed uniformly.

Fig. 14 illustrates a typical application of a Hoffmann bearing, of the double row variety, to a vertical shaft

CUP-AND-CONE BEARINGS

In many cases, more particularly where lightly-loaded bearings are concerned, there is not room to fit a single

or double thrust ball bearing; moreover, the extra weight of thrust or double taper roller bearings is not always permissible in many cases. It is therefore necessary to make some form of compromise by com-

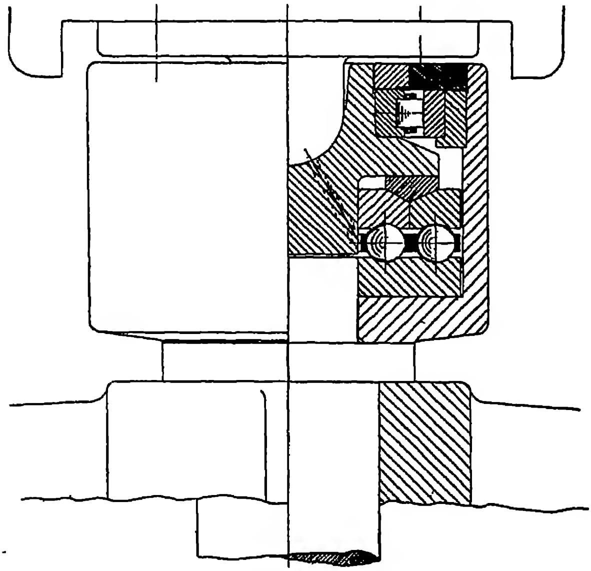


FIG. 14. SHOWING TYPICAL APPLICATION OF DOUBLE ROW THRUST BEARING

bining the journal ball bearing and thrust ball bearing in a single unit Fig. 15 illustrates a bearing of this class, designed to carry both a journal load and to take end thrust in one direction. The bearing shown is known as the *cup-and-cone* type, the inner member being termed the *cone*, and the outer one the *cup*. It gives a two-point contact for the balls, and the radii of the races are rather greater than those of the balls. As a general rule, the radius of each curved race (shown

at R, Fig. 15) is about 10 to 15 per cent greater than that of the balls, i.e. about $1\frac{1}{8}$ times the ball radius. The points of contact of the balls and their races should lie on a line passing through the centre of a ball and inclined to the axis at 25 deg.

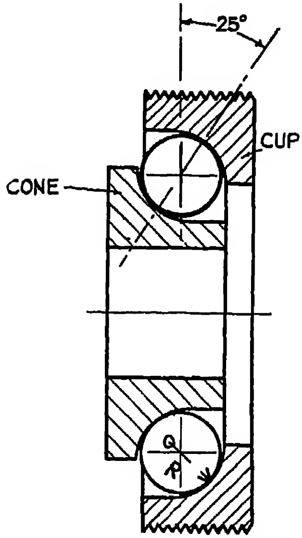


FIG. 15 THE CUP-AND-CONE BEARING

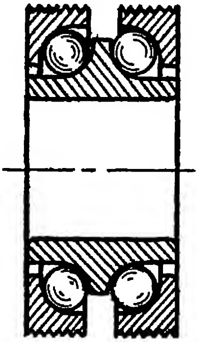


FIG. 16 THIS BEARING TAKES THRUST IN BOTH DIRECTIONS

As the engineer may have to make this type of journal thrust bearing, it may be mentioned that the races are usually made of a case-hardened nickel steel, or a high carbon steel tempered to straw colour and then finished by grinding. There is no cage for the balls in the case of pram and cycle bearings, and it is usual to insert a sufficient number of balls just to fill the space between the cup and cone, allowing a small amount of clearance between the balls to reduce the rubbing tendency of individual balls.

One or both of the races is threaded with a fine-pitch screw thread cut in the lathe, dead true with the cone or cup. It is important to provide adequate means for locking the races on their shafts or in their housings.

In the case of cycle-type bearings the cones are usually turned solid with the fixed wheel spindle, and the cups are threaded externally and screw into the

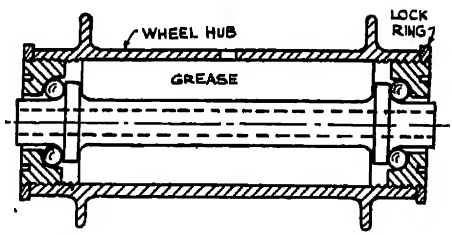


FIG 17. APPLICATION OF CUP-AND-CONE BEARINGS TO MOTOR-CYCLE FRONT HUBS

wheel (or pedal) hub A screwed locking ring is employed to lock the position of the hub.

In connection with the use of cup-and-cone bearings it is usual to fit these in opposing pairs so as to take thrust in both directions, as shown in Fig 16.

Apart from the special examples of cup-and-cone bearings used in various machines, instruments and cycles, there are *standard self-contained bearings* of this type on the market suitable for fitting over turned shafts or in cylindrical housings, just as in the case of journal ball bearings. Examples of these are the Hoffmann combined radial and thrust and New Departure "Radax" ball bearing. The former are supplied to suit shafts of $\frac{1}{2}$ in up to 3 in diameter, and the latter for diameters of 20 mm (·787 in) to 100 mm (3·94 in) An advantage of this type is that it can be adjusted for wear by moving one of the races relatively to the other, for example, the cone towards the cup.

The New Departure "Radax" bearing* illustrated (Fig 17B) is a high-duty non-separable combined radial and end-load carrier, and is, therefore, provided with an angular line of contact (viz. at 15 deg.), as in the case of the cup-and-cone bearing illustrated in Fig. 15. These bearings having plain bore cones and cylindrical outer races, can be located and fitted with ease, they are interchangeable with all standard international sizes of single row journal bearings These bearings have an outer race of high carbon, high chrome steel, whilst the inner race is of high carbon chrome steel; the races are suitably heat-treated after machining, and are ground and polished finally. The standard cup-and cone bearings do not usually have such a pronounced inclination of the ball contact line as the example shown in Fig. 15, the inclination being about 15 deg

FIG 17A HOFFMANN COMBINED RADIAL AND ONE-DIRECTION THRUST BEARING

In the case of the Hoffmann type bearing the balls are held in a yellow anti-friction cage made as a single pressing, whilst in the case of the New Departure a single-piece steel cage is employed These bearings should be mounted so that the heavy shoulder on the outer race and that on the inner race oppose the thrust load

MAGNETO TYPE BALL BEARINGS

A special type of light ball bearing is supplied by most ball bearing manufacturers for use on the armature

* Messrs Delco Hyatt, Ltd, London

shafts of magnetos and on battery ignition contact-breaker shafts. These bearings take only light loads, and are used in cases similar to that of the magneto where the inner race has to be quickly and readily removed from the outer race and housing. They are

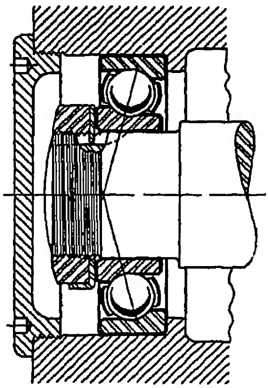


FIG. 17B. THE "RADAX" COMBINED JOURNAL AND THRUST BEARING

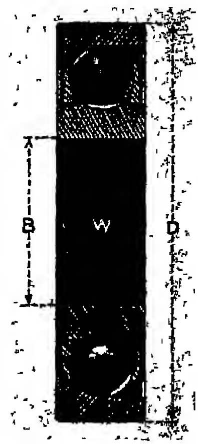


FIG 17C. LIGHT MAGNETO TYPE BEARING (HOFFMANN)

also used for very high speeds, such as for polishing buffs and grinding spindles.

Magneto type bearings are supplied to suit shaft sizes of 5 mm. up to 19 mm. (in millimetre steps in the case of Hoffmann bearings). These bearings, in order to locate the spindle on which they are mounted, must be fixed in pairs, each bearing preventing end movement in one direction. It is important to ensure that the axial adjustment is not so tight as to cause overload on the balls and races.

The adjusting feature in this case is provided merely

to secure correct fitting of the parts in the first instance and not as a means of taking up wear

The three components of magneto type ball bearings, namely, the two races and the cage with its balls, are usually separable to facilitate fitting of the bearing.

LIGHT JOURNAL BEARINGS FOR ELECTRIC MOTORS, ETC.

Apart from the standard sizes of journal ball bearings, most manufacturers now supply special light journal bearings suitable for small high-speed shafts, such as those of small electric motors, electric fans, spinning spindles, etc. They are made to suit shafts of $\frac{1}{4}$ in. up to about 1 in. in $\frac{1}{8}$ in. steps. The corresponding outside diameters are $\frac{3}{4}$ in. and 2 in. ; the respective widths are $\frac{7}{32}$ in and $\frac{3}{8}$ in.

SELF-ALIGNING JOURNAL BEARINGS

Where more than one ball bearing is used on a shaft it is not always possible to align the bearings correctly, and in cases in which the shaft itself may flex or move slightly in axial direction there would be a big stress on the bearings, in the ordinary way, if plain single row journal ball bearings, rigidly mounted as recommended in ball bearing practice, were employed These bearings would be subjected to undue wearing action, for which they were not designed.

It is necessary, therefore, to employ a special type of ball bearing such that whilst capable of carrying its journal load and acting as a journal bearing, it is able to adjust or to accommodate itself to deflection or misalignment of the shaft to which it is fitted. Two different methods of obtaining this accommodation or self-alignment are illustrated in Figs. 18 and 19, respectively

Fig. 18 shows the Hoffmann self-aligning bearing In this case the inner race is secured firmly to the shaft

whilst the outer race has a spherical surface such that it can slide in a spherical seating in another housing member. In the example shown, the inner race is clamped securely between two sleeves, the right-hand member of which bears against a shoulder on the shaft

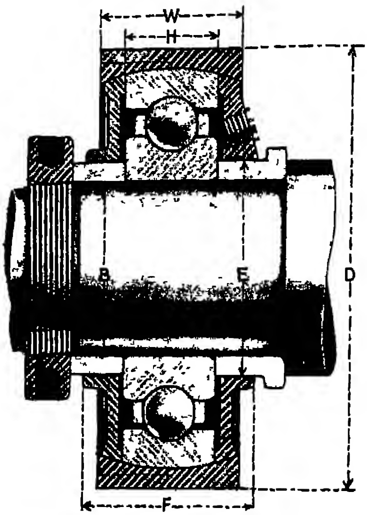


FIG 18 THE HOFFMANN SELF-ALIGNING JOURNAL BALL BEARING

The screwed locking ring on the left clamps the two sleeves together. A dust cover, with grooves in its inner edge, is provided on each side of the two ball races, to keep out dust and water, and to retain the grease injected through the screw hole shown in the right-hand member. Another design of this self-aligning bearing has a split tapered sleeve for clamping securely on to parallel shafts, so that the shoulder in the previous example is not required. These bearings are designed for journal loads only, and it is necessary

that the outer member be free endways to allow it to take up a position opposite the fixed inner race. The shell or housing should accordingly be a nice push fit,

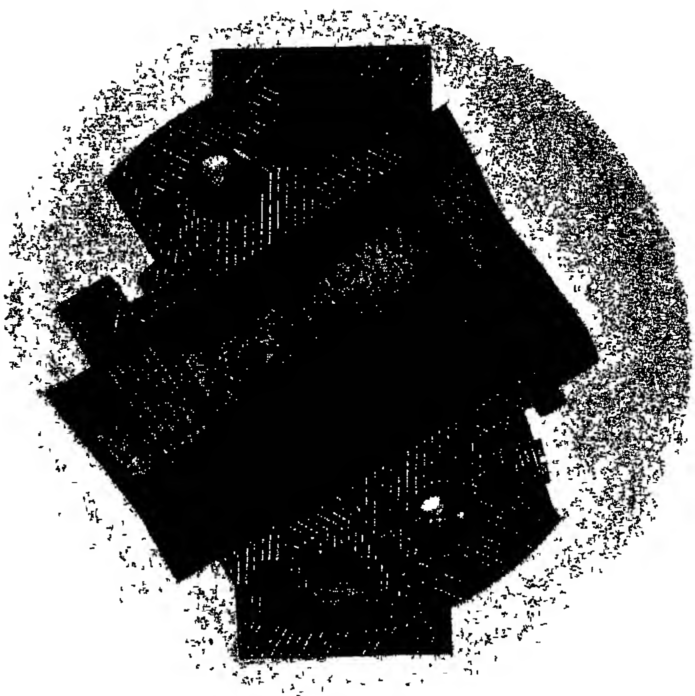
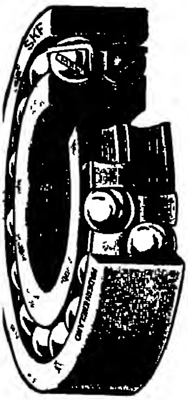


FIG 18A. SHOWING HOW THE HOFFMANN SELF-ALIGNING BALL BEARING CAN ACCOMMODATE ITSELF TO TILTING

free from shake, but not so tight as to prevent the balls moving axially if required. The clearance on either side of this shell should be about one-third of the total width of the bearing.

In the case of the Skefko self-aligning ball bearing

9) the inner surface of the outer ball race is made
erical form, with the centre of the sphere on the
f the shaft. When the latter flexes or distorts, the
simply take up a new position in their spherical
race, and thus in a frictionless and perfectly
manner accommodate themselves to any change



¡ 19 THE SKEFKO
LF-ALIGNING BALL
BEARING

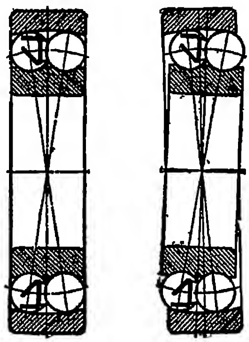


FIG. 19A SHOWING THE PRINCIPLE OF THE SKEFKO SELF-ALIGNING BALL BEARING

ection. As the correction for non-alignment is
ed for in the bearing itself, these bearings require
ecial external members, are self-contained, and
e made to approximately the external dimensions
inary journal ball bearings. In effect, we have in
ype of bearing the ball-and-socket principle It
be observed that there are two rows of balls,
ly spaced by means of a sheet metal cage
-aligning journal bearings are particularly applic-
o line shafting of engineering shops and industrial
ies, to the bearings of overhung shafts, drilling
ine spindles, grinding and polishing machines, for

electrical machines, woodworking machinery, certain automobile parts, textile and paper-making machines, marine work, printing machines, agricultural and mining machinery, and in many other miscellaneous spheres.

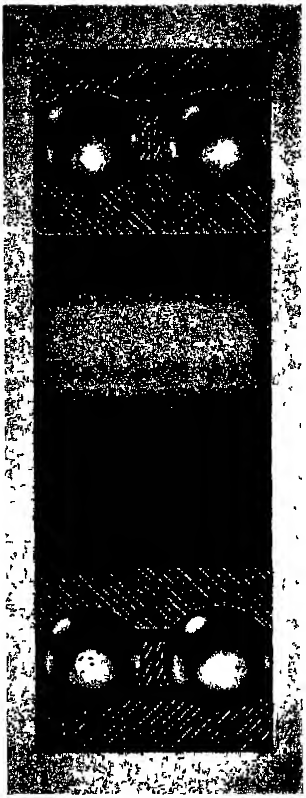


FIG. 20 THE HOFFMANN DOUBLE ROW BEARING FOR HEAVY LOADS

ROLLER BEARINGS

As we have stated, the ball bearing works on the principle of point contact of the balls on their race grooves, and each size of bearing will only safely work with a given maximum load. Where heavier loads and shocks have to be provided for, there are two alternatives to the single row journal ball bearing, as follows: (1) The Double Row Ball Bearing (Figs. 19 and 20), or (2) The Roller Bearing. The latter type gives line contact to the rollers on their races, and so is able to bear much heavier loads than the single or double ball type journal bearings. In the past the design of roller bearings has been attended with greater difficulties than in the case of ball bearings, for troubles were experienced in connection with keeping the axes of rotation, or rolling of the rollers, strictly parallel with

the shaft axis, and in preventing undue end thrust of the rollers

Roller bearings are now available in two principa types, namely, the plain parallel roller journal bearing and the tapered roller bearing The former is usually unable to deal with end thrust whereas the latter can take thrust in one or both directions in addition to journal loads Roller bearings are made also in the *self-aligning* as well as the plain journal or journal thrust types

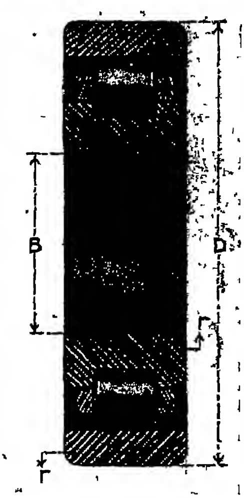


Fig 21 The Hoffmann Plain Roller Bearing

Fig. 21 illustrates the Hoffmann plain roller journal bearing, in which, in place of balls, there is a row of short rollers each of length equal to its diameter. This type will carry a load from 50 to 70 per cent greater than a ball bearing of the same dimensions It will not, however, take any end thrust so that some means must be provided for end location either by plain thrust washers—in the case of light thrusts—or ball thrust bearings In many cases the manufacturers recommend the use of a "*location*" ball bearing of the journal type ; these are similar to standard ball bearings, but slightly smaller in outside diameter. They are mounted immediately alongside the roller bearing, the housing being bored straight through to suit the outer race.

SOME IMPORTANT MOUNTING PRINCIPLES

To obtain the most satisfactory service from ball and roller bearings these must be mounted correctly in

relation to the fixed and moving members. A few typical illustrations will at once demonstrate the principles to be followed in designing the housings for these bearings

Taking the simplest case met with in engineering workshop practice first, namely, that of a rotating shaft supported by and running in two fixed journal ball bearings (Fig. 22, *A*), the mountings must be accurately aligned, so that their axes are coincident. One of the bearings is used to hold, position, or locate the shaft, in this case it is the left-hand bearing. The fixed outer race in this case is a sliding fit in the housing, and it is clamped securely on both sides by the housing, so that it cannot move in an axial direction.

In this method of mounting, as some allowance must be made for end play and expansion, it is usual to allow the outer race of one bearing (in this case the right-hand one) freedom to slide a little each way in its housing, the method of accomplishing this will be clear from Fig. 22, *A*, (right-hand member). The inner races in both cases are interference fits on the shaft, whilst the outer races are sliding fits.

Fig 22, *B*, shows the same example of a two-bearing shaft, but with a roller bearing in place of the right-hand ball bearing, this enables it to take a greater journal load at this end As the rollers are not constrained in any way on the outer cylindrical race, there is no need to allow for any end play in this member as in *A*, Fig 22

In connection with the clamping of the inner races it should be noted that the clamping surfaces do not abut against the sides of the roller track to a greater diameter than the bottom of the roller track, the clamping nuts being recessed for this purpose If the shaft of the two previous examples is subjected to a side thrust in either direction, and this thrust is too great

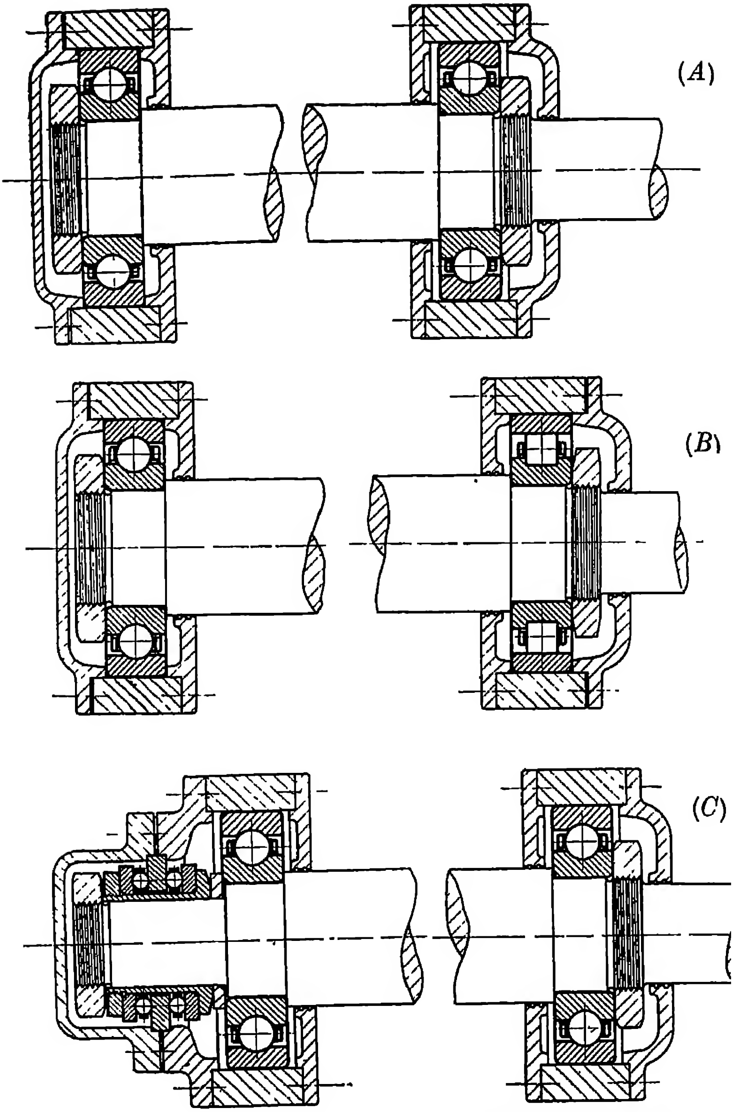


FIG. 22. SOME TYPICAL METHODS OF MOUNTING SHAFTS IN BALL BEARINGS

to be taken by the journal ball bearing, a thrust washer should be arranged in the housing of the fixed or locating end of the shaft.

Fig. 22, *C*, shows how this is carried out. It will be noticed that since the double thrust bearing now locates the shaft, the left as well as the right ball bearing outer race is allowed end play in its housing The Hoffmann

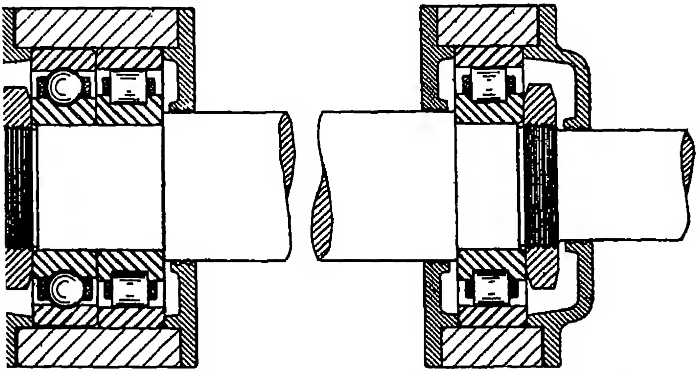


FIG 23. SHOWING THE USE OF A LOCATING BALL BEARING

hrust bearing shown is a complete unit on a sleeve, orrectly adjusted for use by the manufacturers, he sleeve is a push fit on the shaft. If instead of ournal ball bearings a pair of roller bearings has to be sed on account of heavier journal loads, there is no eed to allow any freedom for end movement of the uter races, so that these may be fixed endwise in their ousings; any end movement that may be necessitated is allowed for by the sliding freedom of the rollers n their outer cylindrical races.

Fig. 23 illustrates another important example that equently occurs in practice, namely, of a heavily aded rotating shaft on journal cylindrical roller earings, but which experiences very little or no end

thrust In this case it will be evident that the roller bearings require some means for locating the shaft, as they will not take any end thrust The simplest method is to employ a journal ball bearing to locate the shaft. the ball bearing used is made slightly

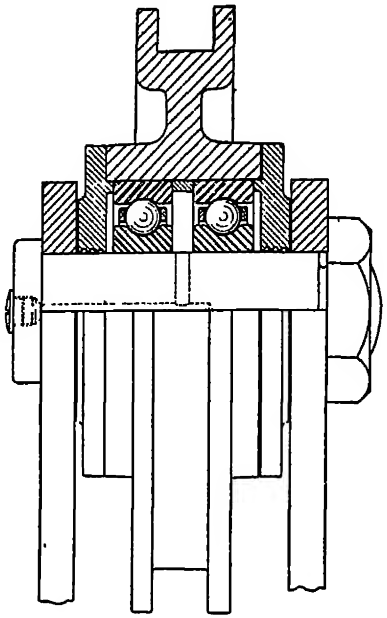


FIG 24 APPLICATION OF BALL BEARING TO A PULLEY WHEEL

smaller than the standard bearing on its outside diameter, and is known as a "locating bearing" It will be observed that the outer race is clear of the housing on its outside surface, but is clamped between the housing cover and the outer roller bearing race, its inner race is an interference fit on the shaft.

Another typical application of ball bearings is

ıllustrated in Fig. 24, namely, that of a pulley wheel running on a fixed shaft. In this case the outer races rotate, and are therefore clamped, with a suitable

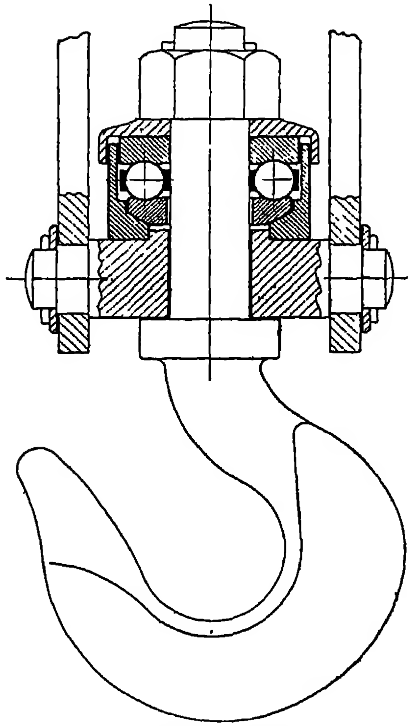


FIG 25. EXAMPLE OF THRUST BEARING APPLIED TO A CRANE HOOK

lıstance pıece between, solıd insıde the pulley. The nner races are a slıdıng fit on the pın or shaft, whılst he outer races are an ınterference fit in the pulley The ›ulley is located by arranging a running clearance ›etween the external faces of the end covers and the

side plates shown. The method of protecting the greas packed bearings with the close-fitting grooved end cov plates should be noted in this example. Fig. 26 shov the application of ball bearings to the loose pulley of

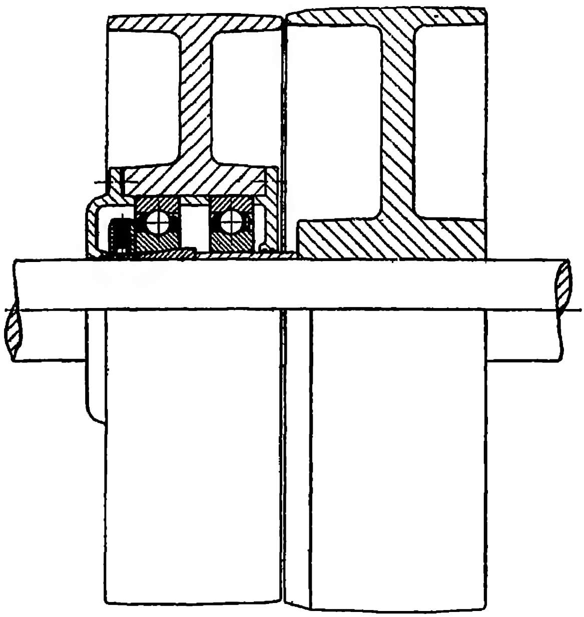


FIG. 26. FAST AND LOOSE PULLEYS SHOWING TAPERED SLEEVE BALL BEARING HOUSING

countershaft. Fig 25 illustrates a good example of tl use of a ball thrust bearing (Hoffmann) in the case of large crane hook. A somewhat similar arrangement used for wire rope shackles and deep-well boring. I this case the shank of the hook is made with a runnir clearance in the bridge piece, and concentric with tl bore of the latter should be a spigot centring tl housing of the thrust bearing. The lower face of tl

nut should run true, between it and the top race of the thrust bearing is a dished cover to keep out dirt and water. It will be noticed that the nut is tightened on its thread until the shoulder of the shank is just

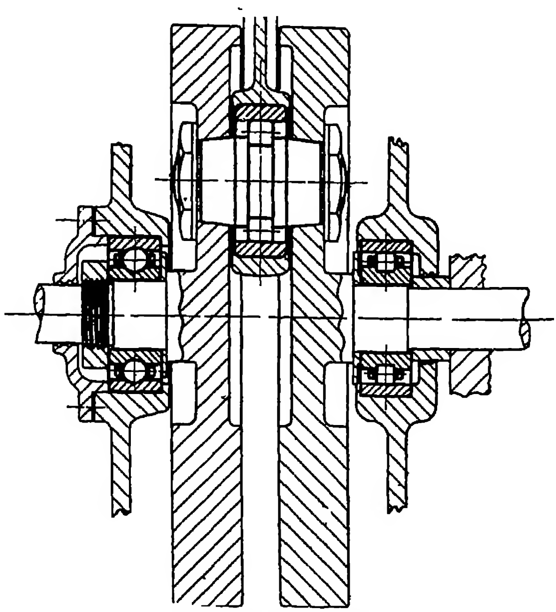


Fig. 27. Showing Ball and Roller Bearings applied to Motor-cycle Engine

lear of the lower surface of the bridge piece. The iousing is filled with grease Fig 27 shows how ball ,nd roller bearings are applied to motor-cycle type ngines.

A common example of the application of light ball earings is illustrated in Fig. 28, which shows an electric an in part section. These bearings must run for long eriods in exposed dusty places unattended, so they

should be well packed with grease and protected as shown.

The particular bearings employed in the case illustrated are the Hoffmann "magneto" type, the outer races being detachable. The outer races each take thrust in one direction, and as they are mounted in an

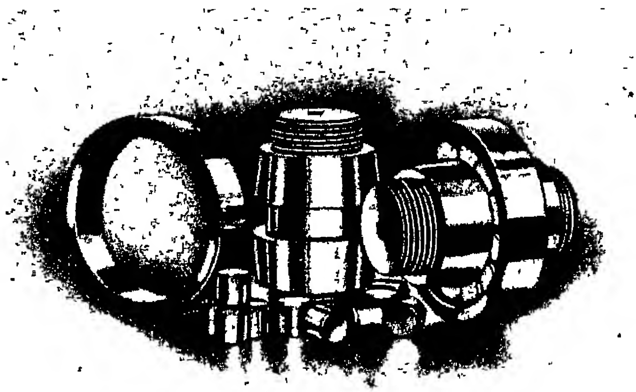


FIG 27A. ROLLER BEARINGS APPLIED TO BIG END OF MOTOR-CYCLE CONNECTING ROD

opposed position, thrust both ways is provided for This arrangement permits the fan to run perfectly in either the vertical or horizontal position. In this case the inner races should be a good fit on the spindle, and must be clamped endwise. One end of the spindle is completely enclosed, the end covers in the other case being bored to a fine running clearance, about ·01 in. larger than the diameter of the distance pieces.

SELF-ALIGNING ROLLER BEARINGS

Fig. 29 illustrates the Hoffmann self-aligning roller bearing which is similar in principle to the Hoffmann self-aligning ball bearing.

The Skefko single row cylindrical roller bearing is illustrated in Fig. 30, whilst the self-aligning type is shown in Fig 30A It will be seen that there are two

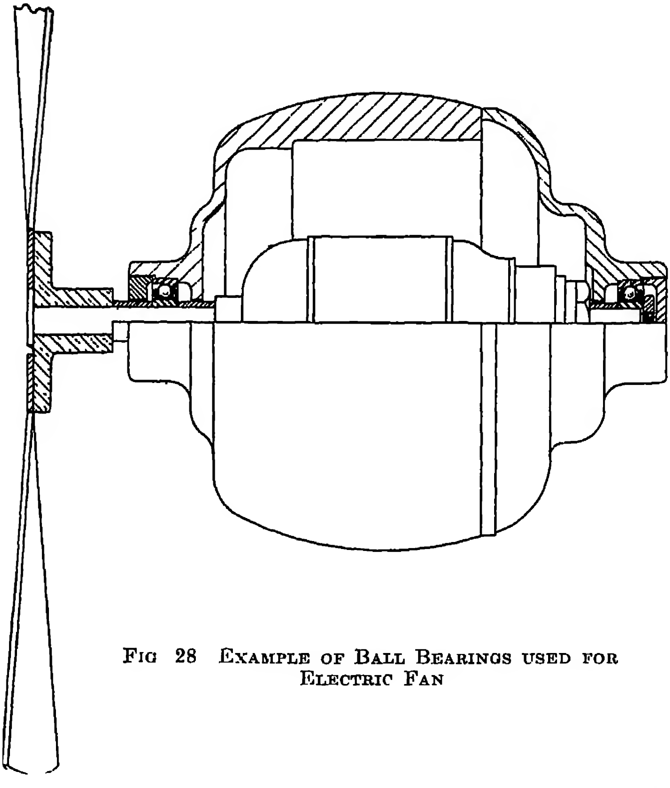


FIG 28 EXAMPLE OF BALL BEARINGS USED FOR ELECTRIC FAN

rows of curvilinear short rollers, which roll on the spherical outer race in such a manner that they can adapt themselves to flexure of the shaft, as in the Skefko self-aligning ball bearing. The Skefko self-aligning roller bearing may be regarded as a heavy duty bearing with self-adjusting properties, capable of

withstanding working loads of about twice the value of those carried by ball bearings of the self-aligning type. These bearings are supplied to suit shafts varying from 50 mm. (about 2 in.) up to 200 mm. (about 7·9 in.).

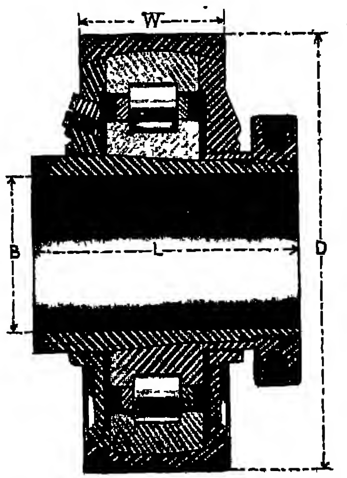


FIG. 29 THE HOFFMANN SELF-ALIGNING ROLLER BEARING

This bearing embodies the same principle as that of the self-aligning ball bearing (see Fig 18A)

Where the journal roller bearings have to withstand also an end thrust in one direction only, it is usual to employ special types of roller bearing having tapered rollers and conical surfaced races for the rollers to run upon.

THE TIMKEN ROLLER BEARING

Figs. 31, 32, and 33 illustrate what is probably the best-known and most widely used taper roller bearing,

namely, the Timken. In this case there is a series of hardened steel tapered rollers, spaced apart by means of

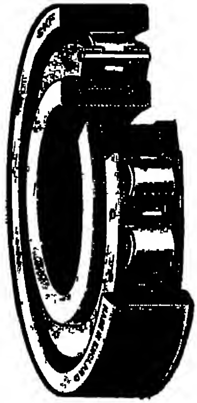


Fig. 30 The Skefko Single Row Roller Bearing

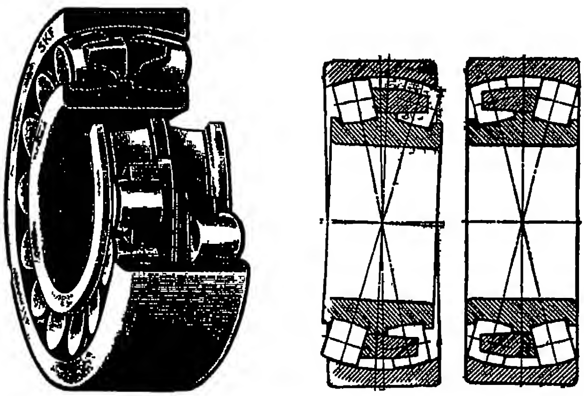


Fig 30a. The Skefko Double Row Roller Self-aligning Bearing

a specially designed cage, having roll pockets punched by a process that obviates distortion. The cage ribs

are winged so that they conform to the curvature and taper of the roll The Timken cage thus provides a pocketing for the rolls which is accurate, thus allowing each of the rollers, without interference, to perform an exactly equal amount of work

The principle of the Timken taper roller bearing is

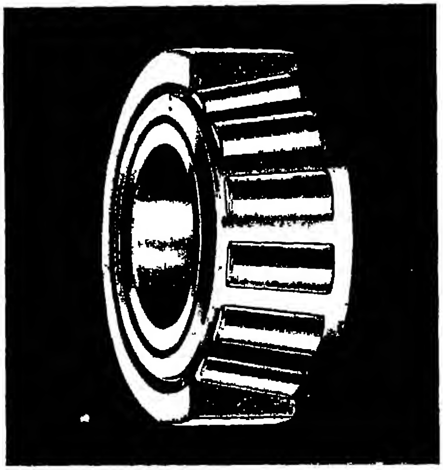


Fig. 31 The Timken Taper Roller Bearing

illustrated clearly in Fig. 33. It will be observed that the races and rollers are all portions of conical surfaces, the apices or conical points meeting at a fixed point on the axis of the shaft to which the bearing is fitted This is the essential condition for pure rolling to be obtained without sliding in any tapered roller type of bearing

In reference to this type, it is of interest to mention that the *inner race* is termed the *cone*, and the *outer race* the *cup*. The shape of the cone should be noted carefully, for it has a raised rim or rib at each side, the

ɔject of which is to provide normal contact areas for ıe rollers. The roller has at its larger diameter end a ıt surface which operates against the rib of the cone his flat surface at the end ' the roller is made to ˌuare exactly with the cen- e line of the roller itself. s each roller revolves ɔout the cone, a normal ·ntact is made against the ne rib in two areas This ntact compels the revol- ng roller to align itself rrectly under all running nditions, and ensures that full-line contact is main- ıned constantly between e roller and the cup on e one hand, and the roller d the cone on the other.

Timken bearings are ıde in a wide variety of es and load capacities for merous purposes. The es vary from those small ɔugh for sewing machines, vn mowers, and small electric fans, up to large ones ployed to withstand exceedingly high thrusts and lial loads, as in steel rolling mills. Between these ne the many intermediate sizes found in line shaft ıgers, lathes, drills, presses, grinders, planers, roll con- ⁊ors, hoisting pulleys, and numerous applications in ı motor-car, commercial vehicle, and tractor Another re recent application is in connection with railway riage axle bearings, to replace the ordinary plain tal bearings, for tests have shown convincingly that

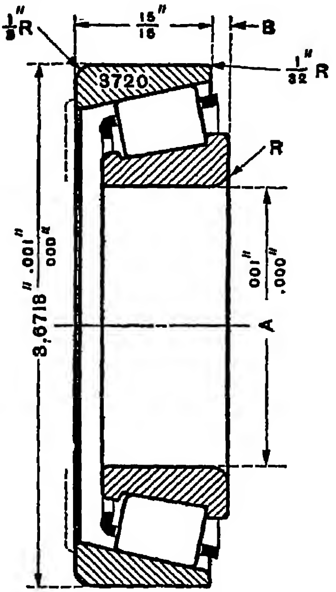


FIG. 32. A TYPICAL TIMKEN BEARING (SHOWN TO SCALE)

a considerable reduction in axle friction is obtained with roller bearings.

Some typical applications of these bearings are given in Figs 35, 36, and 37. The advantage of employing

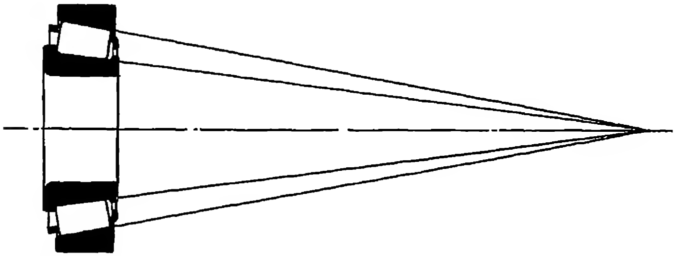


FIG. 33. SHOWING THE PRINCIPLE OF THE TIMKEN ROLLER BEARING

tapered roller bearings is that any wear occurring after long periods of running can be taken up by a slight axial movement of outer race or cup, where the inner race or cone revolves with the shaft

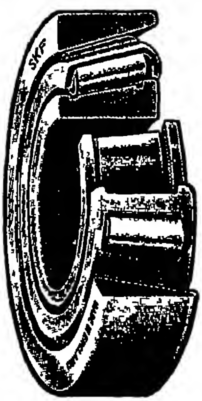


FIG. 34
THE SKEFKO TAPER ROLLER BEARING

THE MOUNTING OF TAPER ROLLER BEARINGS

The following information relates to the mounting of Timken bearings of the type just described. It will be evident from what has been stated that the best disposition is to arrange the bearings in pairs, either with the conical surfaces converging inwards, as shown in Fig. 38 (above), and known as the "*indirect*" method, or outwards, as in Fig. 38 (below) and known as the "*direct*" method. The former method is practically

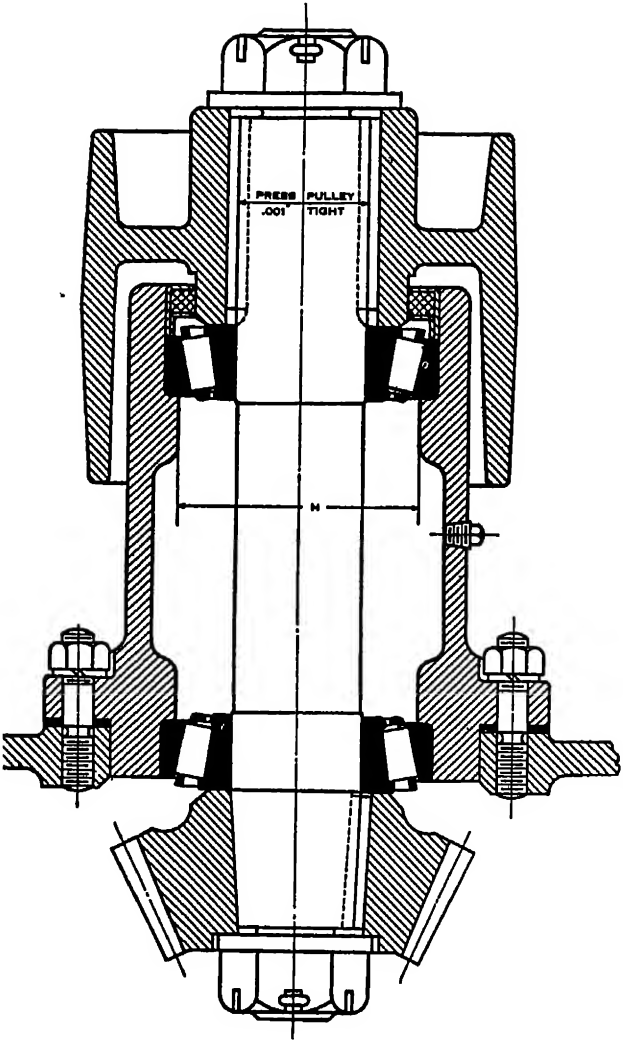


FIG. 35. APPLICATION OF TIMKEN BEARINGS TO PULLEY SHAFT

standard in all stationary shaft applications. A good example is that of the front axle of automobiles, the stub axles housing the fixed races or cones, and the moving wheel hubs the cups. Other examples of indirect mountings include farm implement wheels, conveyor idler pulleys, mine car wheels, overhead

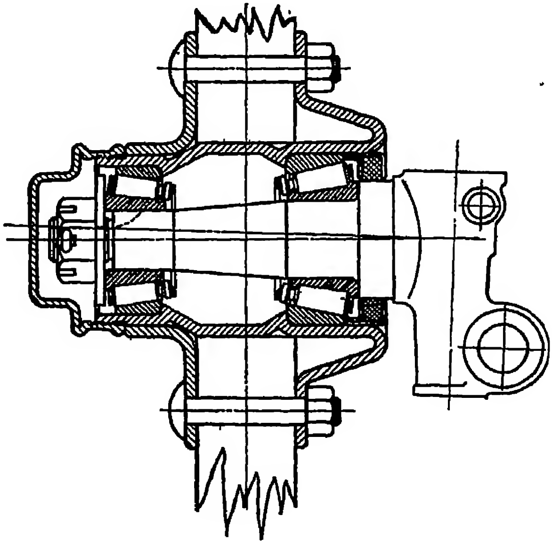


FIG. 36. APPLICATION OF TAPERED ROLLER BEARING TO FRONT AXLE OF CAR

trolley wheels, logging blocks, sheave wheels, hoists, drums, crane wheels, and all similar dead shaft applications. Experience has proved this type of mounting superior to the direct mounting one for wheel or pulley service.

In indirect mounting the distance between the bearing centres should be not less than 15 per cent of wheel outside diameter, and preferably 20 per cent

Adjustable bearing cones with light fits are generally used with the indirect mounting on stationary shafts for speeds below 1000 rev. per min. Slight cone creepage, which results from light fits, is quite permissible on

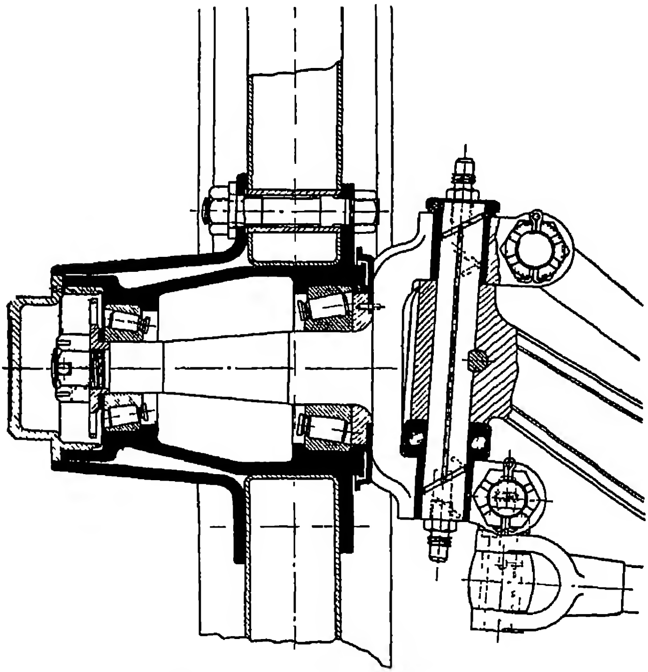


FIG. 36A. ANOTHER FRONT AXLE EXAMPLE

stationary shafts at these speeds Above 1000 rev per min. the indirect mounting and adjustable cones are used with tighter cone fits , in this case cone creepage is undesirable, and must be guarded against by tighter fits.

The manufacturers give tabulated particulars of the sizes, loads, speeds, and fits of their different bearings.

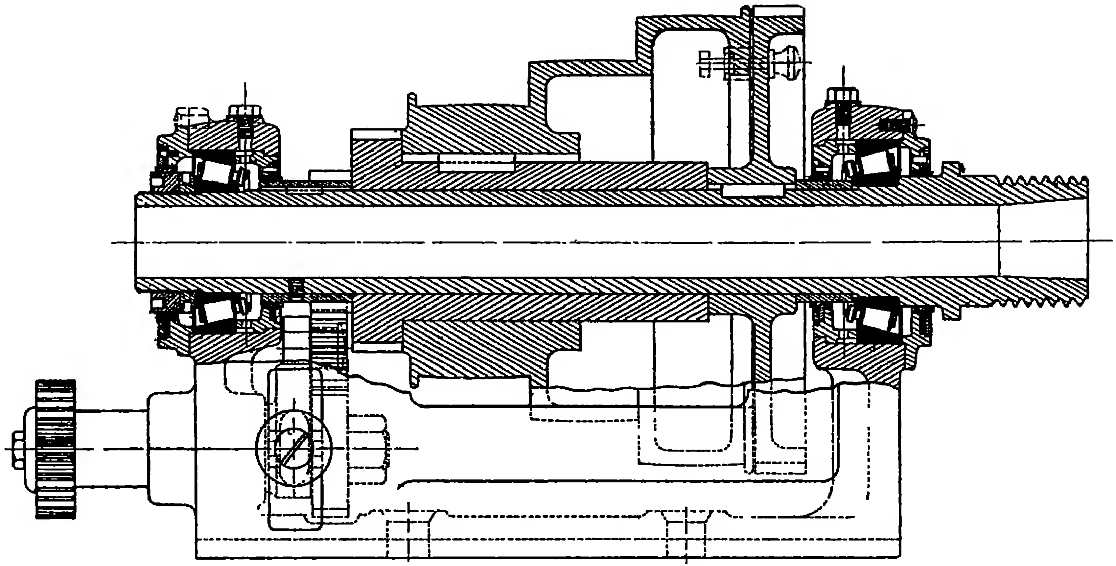
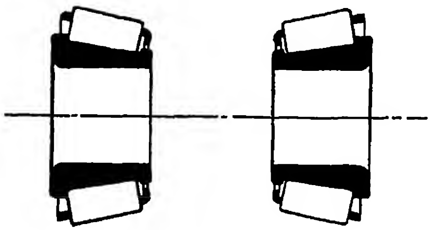


Fig. 37. Timken Roller Bearings applied to a Lathe

THE DIRECT SYSTEM OF MOUNTING

The direct system of mounting (Fig. 38) is used ın
›plications with relatively long bearing centres. It ıs
ore satisfactory above 1000 rev. per min. than the



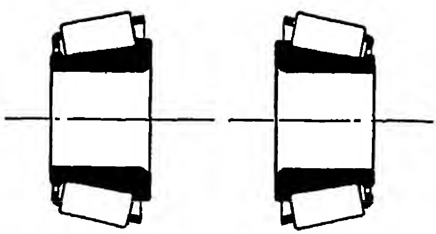


FIG. 38. METHODS OF MOUNTING ROLLER BEARINGS

lirect mountıng, because the cones can be given the
,ht press fits requıred at these speeds. Thıs method ıs
ed widely on automobıle transmission gear shafts,
ferential gear shafts, and lıve axles as well as machıne
ol transmission shafts, for general gear reductions,
low blocks journal boxes, and other applications
ıch permit large bearing spans.

MACHINING PARTS FOR TAKING ROLLER BEARINGS

Referring to Fig. 39, which shows the principal dimensions of the shaft and housing for Timken roller bearings, it is very important to ensure that the

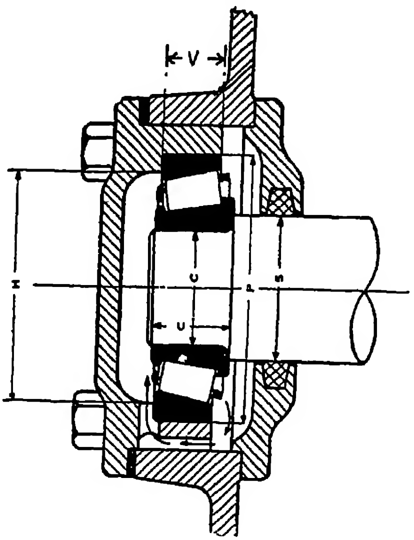
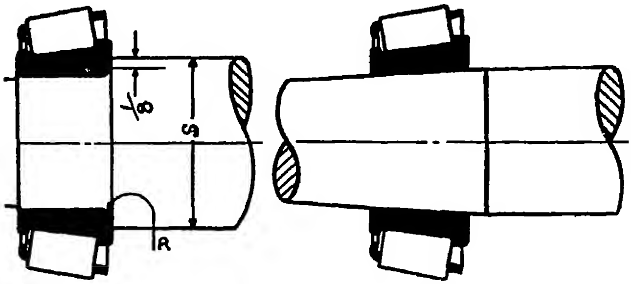


Fig 39. Illustrating Machining Dimensions of Parts for Taking Timken Bearings

machine work is carried out within specified limits to obtain the best working conditions In regard to the "width" dimensions, *U* and *V* (Fig 39), width tolerances of ·020 in. to ·030 in can be permitted in the machine work at each bearing by making provision for this variation in bearing adjustment.

Machine work for two opposed bearings may have total tolerances of ·040 in to ·060 in and be speeded

 the use of machine stops to gauge shaft shoulder
 measurements.
shaft diameter C (Fig 39) tolerance dimension
 39) varying from ·0006 in. to ·0015 in according
aring bore is standard practice for cone seats. The
tion allowable in the housing bore dimension P is
 ·001 in to ·002 in for non-adjusting cups with
 press fits, and from ·0006 in. to ·0014 in. for



'IGS 40A AND 40B TWO METHODS OF CONE HOUSING

ting cups with light press fits. It is absolutely
ial to ensure correct alignment of the cone and
eats

ere possible, the cone should be located on its
by means of a shoulder turned on the shaft $\frac{1}{8}$ in.
r than the radius R (Fig. 40A), the fillet radius
d be $\frac{1}{64}$ in. less than the cone radius The shaft
eter S for all standard cones should be estimated
the dimensions given in the maker's catalogues.
some cases tapered shafts (Fig 40B) are employed
he cone housing. In these instances special
ed bore cones are obtainable This type of fixing
iates shoulders, and is used on live rear axles of
nobiles.

ypical example of the application of a pair of
ct Timken roller bearings to a belt pulley-driven

shaft, having a bevel pinion gear, such as may be found on many machine tools, is shown in Fig. 35. In this case the cups are located by means of recesses bored in the outer housing, the shoulder indicated by the dimension *H* (Fig. 35) varies in height with the cup diameter, section, and radius Adjustment for wear in the bearings can be made by tightening up one or both of the shaft nuts.

ADJUSTMENT OF TIMKEN BEARINGS

As we have mentioned previously, adjustment of this type of bearing is a relatively easy matter, and is accomplished by moving either the cup or cone (whichever is the stationary member) in an axial direction. This feature enables wider tolerances for machining bearing seats to be employed, and makes possible the adjustment of other dependent parts, such as spiral bevel gears, helical gears, and the front axle bearings of automobiles. Further, it is possible by this adjustment to eliminate chatter on machine tool spindles, and generally to reduce noise.

General applications should have ·002 in to ·006 in. end play. End movement of either the stationary or the revolving race, properly fitted, can be used to adjust the bearings for speeds up to 1000 rev. per min. Above 1000 rev per min. it is considered advisable to make adjustments with the stationary race in order to permit a tight press fit of the rotating race. It is recommended that the adjustment increments should not be greater than ·005 in. In the case of two opposed bearings a total end movement of $\frac{1}{32}$ in. should be arranged for. This amount should allow for and include the variations due to machining tolerances, variations in bearing widths, and subsequent adjustments of the bearings. *Threaded adjusting nuts* are widely used for either the cup or cone movement. *Light or medium*

duty applications should use nuts having *sixteen threads per in* , and six or more locking castellations *Heavy duty* machinery may require nuts with *twelve threads per in.* and eight castellations. The threads should be close fitting. Split nuts or housings that can be clamped on mating threads are essential on *extra*

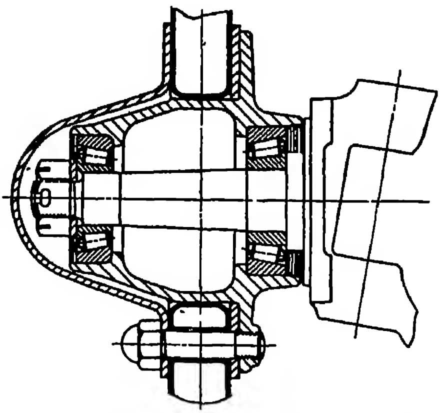


FIG. 41. SKEFKO BEARINGS IN LIGHT CAR FRONT WHEEL HUB

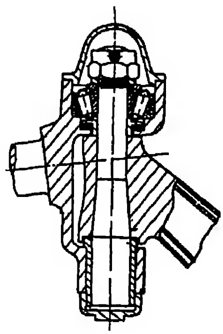


FIG 42. STEERING SWIVEL PIN BEARING

heavy jobs. Threaded adjustments should not only be positively locked against rotation but, wherever possible, the threads should be clamped *to ensure tightness against end play.*

SOME OTHER ROLLER BEARINGS

The Skefko taper roller bearing that is used for many automobile and engineering parts is illustrated in Fig 34 In this case the cone is provided with two grooves on its conical face, and the rollers are housed in a pressed metal cage.

Two interesting applications of this type of bearing are shown in Figs. 41 and 42. Fig 41 shows how a pair

of these bearings is applied to a light car front axle. The design illustrated enables the complete hub to be produced at a low cost, as all screwing is eliminated and accurate machining reduced to a minimum. Fig 42 shows one of these bearings employed to take the weight of the front part of the car upon the steering pin bearing. In this case it takes the thrust (or vertical load) in one direction only and acts as a journal bearing also. The lower end of the steering pivot pin works in a plain phosphor bronze bush housed in the lower end of the stub axle. Provision is made for taking up end wear by means of the castle nut and washer shown above the Skefko bearing.

THE HYATT FLEXIBLE ROLLER BEARING

Another widely used roller bearing is the Hyatt type illustrated in Figs. 43 and 44. The principle of this bearing is quite different from that of any others we have described, for instead of employing solid ones in the bearing, the Hyatt type utilizes flexible rollers. These enable the bearings to accommodate themselves to small errors in alignment, and enable the rollers to bear evenly all along their tracks. It is claimed also that this type of bearing, on account of the resiliency of the rollers, will withstand impact shocks, and load variation much better than solid roller bearings.

The rollers are in the form of hollow cylinders formed by helically winding strips of alloy steel. After being cut to length, the rollers are heat treated, not only for maximum hardness, but also to develop maximum toughness; they are then very accurately ground to size. In this way very tough rollers offering maximum resistance to shocks, abrasion, and fatigue are obtained.

In assembly the consecutive rollers are arranged with right- and left-hand spirals (Fig. 44); this is for the purpose of directing the lubricant back and forth

ross the bearing surfaces, so as to ensure continuous
orication of all parts.

The rollers are kept in proper alignment, and are
gularly spaced by means of a cage or retainer. This
nsists of two pressed steel end rings held rigidly in
rallel and concentric position by means of round
el spacing bars, the ends of which are riveted in

43. The Hyatt Flexible Roller Bearing with Slit Outer Race

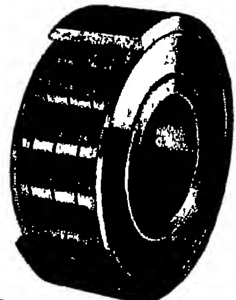


Fig. 44 The Narrow Type Hyatt Bearing

untersunk holes in the end rings. The combination of
llers, cage, and bars is termed the *roller assembly*;
is a unit that contains the rollers and can be handled
parately, since the rollers cannot come out of place.
e length of the roller assembly measured between
tside faces of the end rings is always shorter than the
ter race by a definite operating clearance, so that
t surfaces located directly against the ends of the
ter race will not interfere with the operation of the
aring.

Two kinds of roller assembly are manufactured;
ese are known as the "light" and "heavy" types.
In regard to the design of the *outer race, there are*
o distinct types, known as the *Slit* and *Solid*
spectively

The Slit Outer Race is shown in Fig. 43. It is used chiefly for light duties, the rollers being comparatively long, it is designed primarily for direct roller operation No inner sleeve or race is fitted, the rollers operating direct on the steel journal. A slit outer race is, however, provided; this is of high carbon steel accurately finished to cylindrical form. The joint is in the form of an obtuse angle (Fig 43), so that the rollers operate over it without shock or noise.

Before pushing into its housing the slit race will be found to have a slight outward spring, which causes the joint to open slightly. When brought to true cylindrical form by pushing into the machined cylindrical housing the joint is closed up to its proper width and the spring of the race eliminates any tendency to creep or turn in the bore. *The load-carrying capacity* of this type is governed to a large extent by *the hardness of the journal* upon which the rollers operate. This type of bearing has been widely used for back axle bearings of Ford cars.

The Solid Race Type, Hyatt roller bearing (Fig. 44) is intended for heavy loads and higher speeds. In this case the rollers are fairly short, and both inner and outer solid race are provided

As before, the spiral type of rollers with their cage is employed. The races are made in the form of straight cylinders of steel case-hardened all over to ample depth and ground to close limits.

The three components, namely, the inner and outer, races and the roller assembly are separable, and, if desired, either the inner or outer races can be dispensed with if suitable operating surfaces are substituted This is a very useful feature in engineering work, for an appreciable amount of bearing space can be saved if one of the races can be dispensed with.

The correct size and type of bearing to employ in

particular cases will depend upon the speed, shaft hardness (where no inner race is used), nature of service, and the maximum temperature to which the bearing is exposed. Hyatt bearings will operate on the normal rating basis up to 350° F. Above this temperature the temper of the races is affected, and the load capacity therefore reduced Thus at 600° F. the capacity is reduced to one-third.

The manufacturers supply tabular and graphical data in regard to the appropriate sizes of roller bearings to work under specified conditions

The slit race bearing is obtainable in a range of sizes varying from ½ in. diameter shafts (with corresponding working loads of 62 lb. at 100 rev per min., and 33 lb. at 1000 rev. per min.) up to 5 in. diameter shafts (with corresponding working loads of 5400 lb. at 100 rev. per min. and about 3000 at 500 rev. per min. for bearings of 6 in. width).

HYATT BEARING MOUNTING NOTES

The principal points to observe when designing mountings and housings for Hyatt bearings can be enumerated, briefly, as follows—

1. *Correct alignment of bearings and shafts.*

2. *Retainment of the bearing parts* in their correct operating position by means of suitable collars, shoulders, sleeves, or recesses, as in the case of ball bearings

3. *Endwise location of the shaft* Hyatt roller bearings do not take end loads, so that it is necessary to provide some means to locate the shaft in its correct lengthwise position and to take any end thrust that may occur. For small end thrusts and at low to normal rotational speeds, plain metal thrust washers may be used. For appreciable end thrusts, ball thrust washers are

necessary. Sometimes a deep groove location journal ball bearing can with advantage be used.

4. *Protection of the bearing* As in the case of other ball and roller bearings, some means must be provided to exclude dust and water and to retain the lubricant

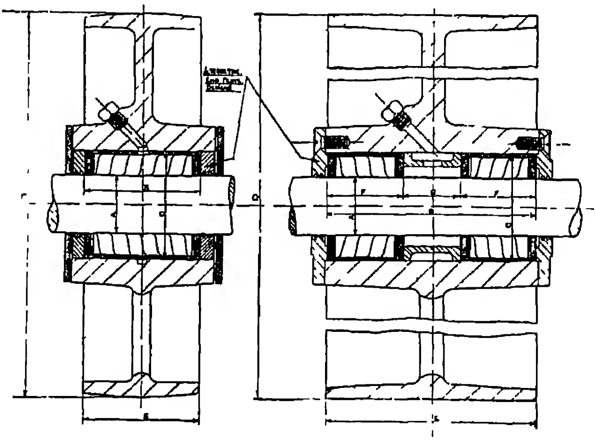


FIG 45 SHOWING APPLICATION OF HYATT BEARINGS TO PULLEYS AND SHAFTS

in the bearing The most common method for a blind end shaft is that of an end cap or plate sealing this end of the bearing

For bearings having shafts passing through, a cover ring having a small clearance around the rotating shaft, and provided with a series of small grease-retaining grooves, is the most commonly used Sometimes, as in the example illustrated in Fig. 46, a felt retaining washer is housed in the cover ring. The illustration referred to shows the arrangement of a railway carriage axle running on two sets of Hyatt bearings, in place of the usual plain gunmetal axle boxes , the friction is

ıus considerably reduced. It should be noted that ıere is a plain metal disc to take the side thrust, and ıat the "open" end of the bearıng has an inserted ·lt washer to retain the lubricant and to exclude dırt ıd water. The complete bearıng is packed wıth grease,

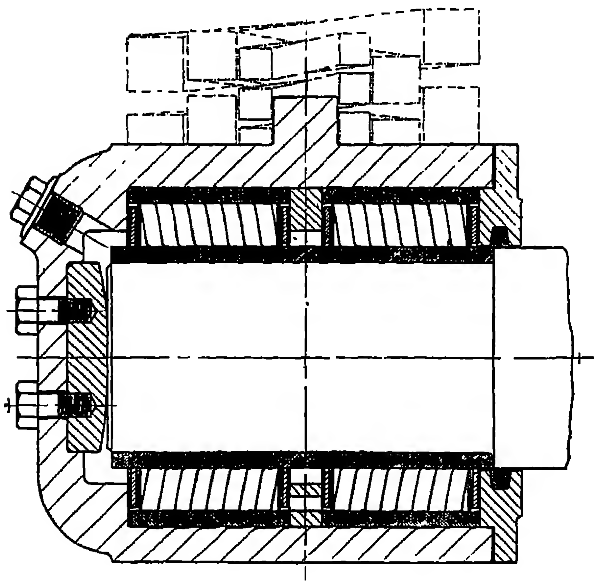


FIG 46 HYATT BEARINGS AS USED FOR RAILWAY CARRIAGE AXLE BEARINGS

ıected or replenıshed through the hole shown wıth ıts ·ewed plug on the left.

Fıg 45 shows another applicatıon of Hyatt roller arings to loose pulleys of machıne drıving shafting ıe left illustration shows a narrow, and the right a de pulley. In the latter example two sets of roller arings, with a rıng spacer, are employed The end

plate protecting members should be noted in these examples

Fig 47 shows the method of employing Hyatt roller

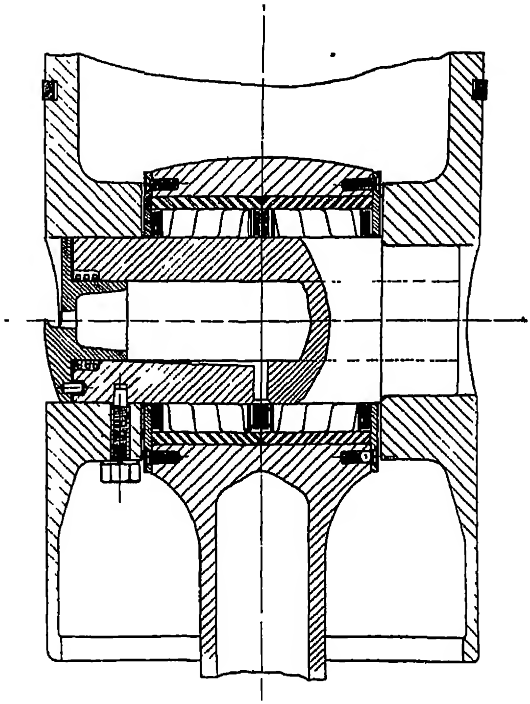


Fig 47. Hyatt Roller Bearings used for Small Ends of Connecting Rods of Large Diesel Engine

bearings for the connecting rod and gudgeon pin bearings of large internal combustion engines.

5. *Lubrication.* As with ball bearings the lubricant is intended, not to reduce friction of the rollers in their races, but to protect the surfaces against corrosion, and

lso to act as a seal to exclude dirt and moisture. The ıbricant does, of course, tend to reduce friction of the age and rollers.

As a general procedure, and in cases where the ıbricant can be properly replenished, *oil* should receive rst consideration ; after this a thin grease, and finally thick grease.

If the temperature of the bearing is high a suitable igh viscosity lubricant should be employed ; usually cylinder oil is used for temperatures of 400° to 600° F. cup grease is recommended for temperatures between 2° and 180° F. The use of greases containing any olid fillers such as cork, asbestos, graphite, or mica hould be avoided, as the solid material is rolled into akes and is harmful to the bearing.

THE LUBRICATION OF BEARINGS

In the cases of both ball and roller bearings, as the ction of the balls or rollers is purely one of rotation and ot of sliding, the question of lubricating the balls or ollers is unnecessary as a means of reducing friction. t is clearly not possible to maintain a film of lubricant etween the ball or roller and its races, owing to the elatively great intensity of pressure.

The principal object, however, in lubricating these earings is to protect the high polished surfaces from orrosion, and to keep out dust and water. Either rease or oil will do for this purpose, provided that it an be kept in the bearing. In the majority of cases it found more convenient to use grease, as this will emain in the bearing for long periods without needing eplenishing. The cages are also lubricated by the oil or rease supplied, to reduce sliding friction. A lubricant he consistency of vaseline, *non-acid in* character, is uitable for most conditions and temperatures up to

120° F. The melting point should not be lower than about 200° F.

In cases where the bearing is exposed to higher temperatures either a high melting point grease or a mineral lubricating oil should be used.

Animal and vegetable greases or oils are apt to develop free acid, and therefore should be avoided in ball and roller bearings.

In many cases oil is preferable to grease, notably where bearings run at high speeds, and may be subjected to higher working temperatures. Petrol engine bearings are a case in point. When ball bearings are used in gear-boxes and certain engine bearings, they are lubricated with oil in preference to grease.

PROTECTION OF THE BEARINGS

Apart from the use of a grease lubricant to keep out water and small dust particles, it is usually necessary to provide additional means for protecting ball and roller bearings from ingress of dust and water, or these agents would quickly cause abrasion and corrosion. Moreover, the lubricant must be kept within the bearing also. It is usual to make provision in the design of the housing or shaft for this protection. One common method is to provide a wide lip to the bore of the end cover and to make it a fairly close fit to the shaft, with a clearance of about $\frac{1}{100}$ in. In the bore a series of grooves are turned (Fig. 48), which become filled with grease and act as a seal, effectively excluding dirt and water. Semicircular grooves of about $\frac{1}{16}$ in diameter are found to give good results.

For particularly dusty conditions the addition of a thrower over the lip of the end cover (Fig 49) is required.

Fig. 49 shows an excellent form of bearing protector, which has proved most satisfactory under conditions similar to those of axles on motor vehicles in excluding

grit and water This combination of a thrower and two grooves is both simple and effective under severe conditions of bearing operation

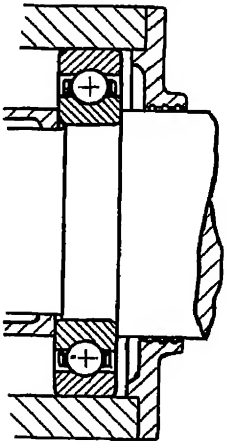


FIG. 48

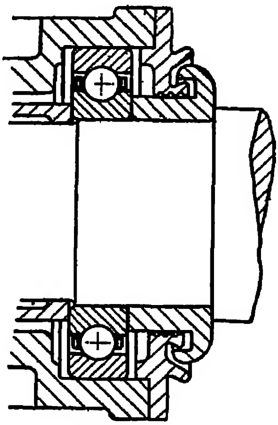


FIG 49

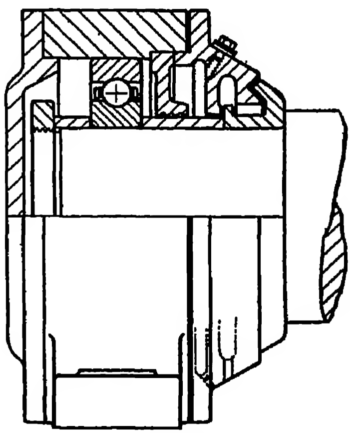
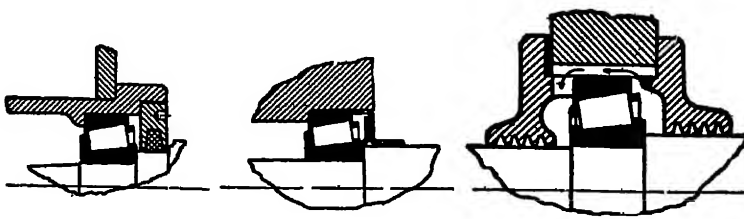


FIG. 50

ILLUSTRATING ALTERNATIVE METHODS OF PROTECTING BEARING ASSEMBLIES

When bearings have to be protected against the force or pressure of water, they should be packed with grease, and a grease-gun used for forcing the grease into place. Many motor-car parts are thus protected. Fig. 50 illustrates a good design of bearing (Hoffmann) protection for such cases.

The worst conditions likely to be met with are those of bearings running totally immersed in water or other



(A) (B) (C)

FIG. 51. ILLUSTRATING THREE DIFFERENT METHODS OF PROTECTING ROLLER (AND BALL) BEARINGS

A = Felt disc *B* = Sheet metal discs *C* = Annular grooves

liquids. In such cases, apart from the adoption of the best available design of housing protection, stainless steel bearings are advisable. Although more costly and of lower load capacity, the non-corrosive property of this steel renders it the most suitable. Fig. 51 illustrates some alternative protective measures or *closures* recommended for Timken roller bearings; these employ respectively (from left to right) felt washers, sheet metal labyrinth discs, and annular grooves.

Other examples of bearing protection will be found among the illustrations of this section.

SECTION XXXII

PETROL ENGINE FITTING AND ASSEMBLY

BY

MAJOR A. GARRARD, Wh.Ex.

SECTION XXXII

PETROL ENGINE FITTING AND ASSEMBLY

INTRODUCTION

Practical knowledge of the fitting and assembly of any engineering product is more readily obtained in a repair shop than in a new construction factory. And this difference holds also in connection with internal combustion engines. In the first place, wear and the other results of use or misuse gives much useful information which serves as a guide when fitting and erecting , and in the second place, there is much more careful and skilful fitting required.

Modern manufacturing methods tend more and more in the direction of perfection and exactness in machine processes In those works which are organized on what are known as production lines, the various components of a petrol engine are not "fitted," but are merely bolted together, and the operative has little opportunity of learning more than one or two operations In a number of British motor-car and motor-cycle factories, the engines are, however, produced by a combination of clever machine processes and accurate and skilful fitting

An endeavour will be made in the following to consider the fitting and assembly of petrol engines from the point of view both of new construction and of repair work, but the latter will necessarily receive somewhat more attention, and reasons will always as far as possible be given in connection with the operations described.

The reader is assumed to have some general acquaintance with the construction and methods of working of petrol engines. Only a few typical examples of engine construction have therefore been described or illustrated.

All available drawings and instruction books relating to the particular makes of engines in which the reader may be interested should be carefully studied.

The greatest care must be given when assembling to ensure absolute cleanliness of all the parts. Grit, sand, emery powder, or any other abrasive will soon ruin an engine. It is difficult to remove completely all the moulder's sand from the corners and inaccessible parts of castings. A sand blast is very penetrating, but may leave a residue in some parts. Probably the most effective way to prevent damage (after all possible sand has been removed by ordinary methods) is to fix the residue by the application of a coat of suitable oil-resisting paint to all the interior surfaces of the casting. This paint may be applied by the brush, but spraying is a quicker and more effective method.

All internal surfaces must be wiped clean, paraffin being used where necessary, and oil should be applied to the bearings, cylinder walls and other working parts before finally bolting up or assembling.

Nuts and studs must be screwed up tightly, but not too tightly. Experience only will enable the exact degree of force to be estimated. All nuts must be prevented from slacking back either by split pins, spring washers, or lock nuts. A missing split pin from, for instance, a big-end bolt may cause serious damage.

ENGINE STANDS

Stationary petrol engines are generally intended to be bolted down to a flat surface, and they are therefore

conveniently supported on the bench or on a raised platform of some kind for assembly or repair purposes.

Petrol engines for motor-cars and motor-cycles, on the other hand, are supported by brackets at their

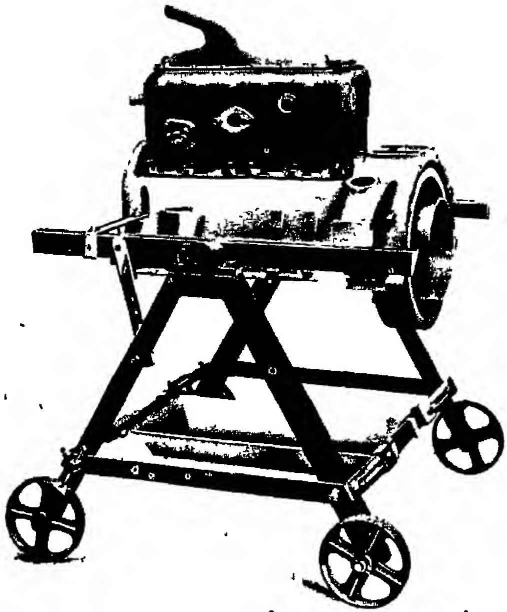


FIG. 1. REVERSIBLE STAND ADJUSTABLE FOR ENGINES OF ALL SIZES

sides or ends, and have oil sumps on other parts below the bracket level. Further, work of any kind is facilitated if the engine can be inverted. Such engines cannot therefore be rested on a flat surface, and special stands are employed wherever efficiency is properly considered.

A good example of a stand adaptable for engines of

all types or sizes is made by Mann, Egerton & Co., Ltd (Fig. 1). The engine may be bolted to side members carried by trunnions on the triangular side frames, the side members having undercut grooves for the securing bolts The engine is held by stays in any desired position, upright or inverted The width of the stand, that is the distance apart of the triangular side frames. is adjustable.

LUBRICATION

Before lubrication systems can be dealt with and the work of fitting bearings can be done with intelligence and understanding, it is necessary to know something of the principles of bearing friction and lubrication

The main principle to bear in mind is that there should never be any actual metallic contact between the bearing surfaces In the case of a big-end or crankshaft bearing, the steel journal and the white metal bearing must be separated by a thin film of oil about $\frac{1}{1000}$ in. in thickness, which in a way may be likened to the rollers in a roller bearing Oil is more or less viscous according to its quality, that is, the particles tend to adhere together; and a thin layer or film between two surfaces will resist enormous pressures before it can be squeezed out, even though its temperature rises in use and it becomes much less viscous than at ordinary temperatures. It is thus clear that all efforts must be directed to maintaining this film of oil, and journals and bearings must be so fitted that their surfaces agree very closely, and leave only the necessary uniform clearance for the oil film

In modern engines, all the working parts are lubricated automatically with the exception of the ball bearings in the electrical accessories, such as the magneto. As the crankshaft and connecting-rod big-end bearings work under the most severe conditions,

ɪrication systems are classified according to the ɪnner in which these parts are supplied with oil.

Ƭhe systems are as follows—

ɩ. Forced feed in which a pump forces oil under ɪssure to the crankshaft bearings, and thence through holes or passages in the crankshaft to the big-end ɪrings.

2. Splash systems in which the oil is pumped into ɪughs, one for each big end, a dipper on the big end tributing the oil.

3. Systems, sometimes called "petroil," in which a inite quantity of oil is mixed with the petrol.

4. Systems in which oil is fed to the crank chamber a steady rate by a hand or automatic pump. A ɪible drip feed facilitates regulation of the rate of flow. ɪis system is largely employed on motor-cycle engines.

Systems 3 and 4 are only applicable to engines having ll or roller bearings. Forced and splash systems are ɪen combined in the same engine ; in engines having erhead camshafts, oil is generally forced to the arings.

There is much variety in design and, when dealing th any particular engine, the general working of the brication system should be mastered by reference to e maker's working instructions, drawings, or other formation.

Whatever system is in use, it is essential that all ɩways, whether formed by external or internal pipes, by passages in the casting or shafts, should be clean ɪd clear. Paraffin is a most useful cleansing agent in is connection, and wire may also be used where ɪcessary.

All pipe joints and connections, whether internal or ɪternal to the engine, must be perfectly oil-tight, as akage at any point may upset the working of the ɪstem.

In all engines with a circulating oil system fine gauze filters are used, generally in such a position that they filter the oil just before it enters the pump. Gauze filters are also employed for the fresh oil supplied to the engine. The greatest care should be taken that all such oil filters are perfectly sound. There must be no defects in the gauze material, and any joints must be

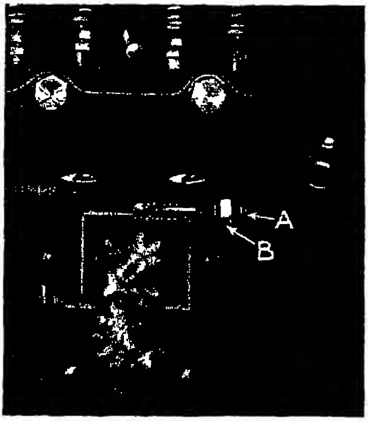
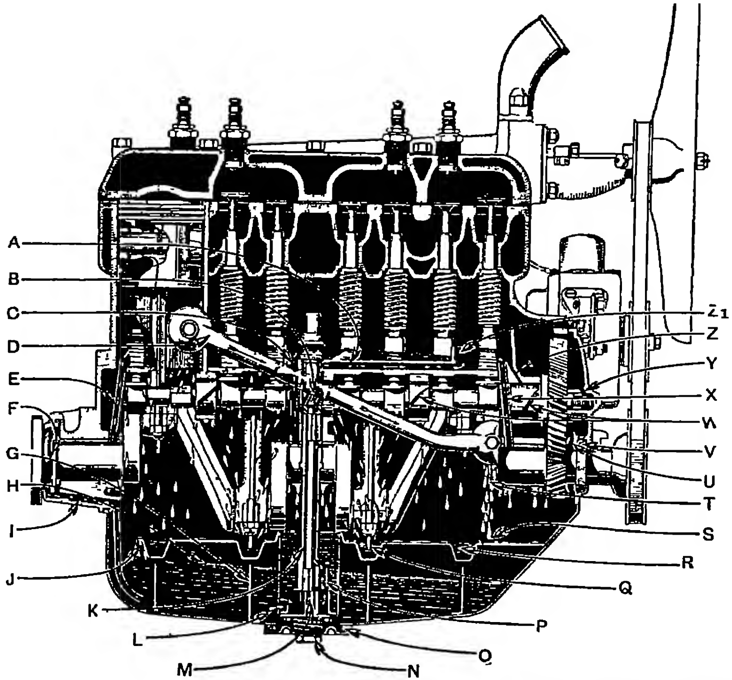


FIG. 2 GEAR-WHEEL OIL PUMP

perfectly made. The soldering iron may be used to correct any defective joints.

Two types of oil pump are in use—the gear wheel pattern and the plunger pump.

In the gear wheel type, it should be noted that the intermeshing teeth do not carry oil either one way or the other; they act merely as a partition. The oil is carried from the suction side to the delivery side by the spaces between the teeth and the casing. These pumps are very reliable, and do not depend for their operation upon springs or valves. Pressure relief valves are often provided, as shown in Fig. 2, in connection



A = Main oil line from pump to valve chamber reservoir
B = Oil pump driven by spiral gear on camshaft
C = Oil line to centre crankshaft bearing
D = Overflow pipe from valve chamber reservoir to oil pan tray
E = Oil line to rear crankshaft bearing
F = Seal prevents oil from escaping into clutch housing
G = Baffle plates prevent surging of oil
H = Main oil reservoir
I = Oil return pipe from crankshaft rear bearing
J = Oil from tray overflows into main reservoir
K = Oil passage to valve chamber reservoir surrounds pump shaft
L = Oil drawn into pump through filtering screen
M = Oil pump gears
N = Oil pan drain plug
O = Oil pump removal cover
P = Gear type oil pump
Q = Connecting rod dipper forces oil into big end bearings and lubricates cylinder walls, pistons, and other moving parts by oil spray
R = Oil trough for connecting rod dipper.
S = Oil pan tray
T = Overflow oil from valve chamber reservoir drains into oil pan tray
U = Oil throw off ring
V = Oil-tight packing on front end of crankshaft
W = Spiral grooves distribute oil full length of camshaft bearing
X = Oil line to front crankshaft bearing
Y = Oil lead to camshaft end thrust plunger
Z = Oil lead to front crankshaft bearing
Z_1 = Valve chamber oil inlet

FIG. 3. FORD FOUR-CYLINDER ENGINE

with the Standard 9 h p. engine This valve passes oil from the delivery side directly back to the suction side when the oil pressure exceeds the limit to which the valve is set by the adjustable screw *A* abutting against the end of the spring. *B* is a lock nut

The plunger pump is generally moved in one direction, that is, the delivery stroke, by a special cam on the camshaft; it is returned by a spring. This type of pump depends entirely upon the automatic opening and closing of one-way valves, one or two of which are provided on the suction and delivery sides. These valves consist of a hard steel ball bearing on a narrow bevelled or coned seat frequently made of bronze. Close engagement of the valve with its seating may be obtained by giving the ball, when it is in position, a few light taps with a hammer, using a short punch of brass or similar soft metal

The plunger will force the oil without leakage if it is a very close fit in the pump cylinder. Shallow grooves turned in the plunger help to prevent leakage. Wear is very slow, but very little wear indeed necessitates a new plunger

LUBRICATION SYSTEMS AND GENERAL CONSTRUCTION

A number of examples of different lubrication systems will now be referred to and illustrated. Constructional characteristics, other than those relating to lubrication, will also be dealt with

In the latest Ford four-cylinder engine, Model AF, shown in Fig. 3, the oil is drawn from the sump through a cylindrical filtering screen into the gear pump, and is discharged into the valve chamber at the front or raised end of the engine. A medium light oil is used, and it is not under pressure The valve chamber is divided into three reservoirs by two small transverse

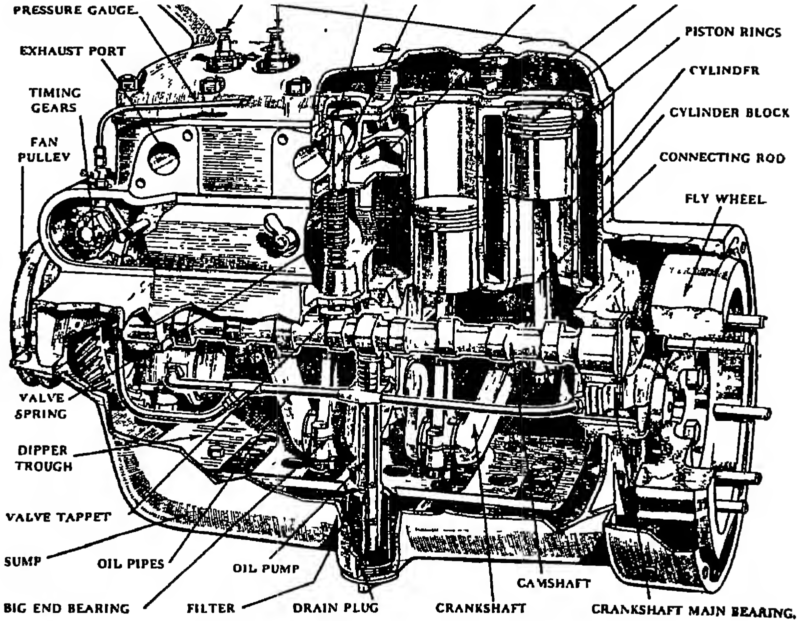


Fig 4. Morris Four-cylinder Engine

dams, and the oil overflows each dam in succession and fills the next lowest reservoir. From the several reservoirs, oil flows by gravity to all the bearings and other working surfaces and to the oil troughs in the oil pan tray.

In the Morris four-cylinder engine, Fig. 4, a vertical plunger pump worked from a cam at the middle of the camshaft draws oil through a filter from the sump, and delivers it under pressure to the three crankshaft bearings by means of the distributing pipe shown. A branch from this pipe also delivers oil to the timing gears. Surplus oil fills the troughs from which the big ends are lubricated. The pressure of the oil delivered to the main bearings is indicated on a gauge.

The Morris six-cylinder engine, Fig. 5, is pressure lubricated throughout. The oil is strained through a horizontal wire gauze covering the whole of the sump. A gear wheel pump driven by a vertical shaft draws oil from the sump through pipes and drilled passages, and delivers it through similar means to the crankshaft bearings and big ends, the spindle of the tensioner for the chain driving the overhead camshaft, the bearings of the camshaft and the rocker arms operating the overhead valves. In this, as in all full pressure systems, the surplus oil escaping from the big ends is relied upon to lubricate the cylinder walls and gudgeon-pin bearings.

The Douglas two-cylinder-opposed, air-cooled motorcycle engine is shown partly in section in Fig. 6, the inlet valves being arranged at the top and the exhaust valves at the bottom. The exhaust valve lifter, which reduces the compression pressure and thus facilitates starting, is arranged at the bottom of the crankcase. A bell-crank lever is operated by a Bowden wire, and the short arm lifts the valve tappet slightly. Fig. 7 shows a diagram of the drip-feed, pressure oiling

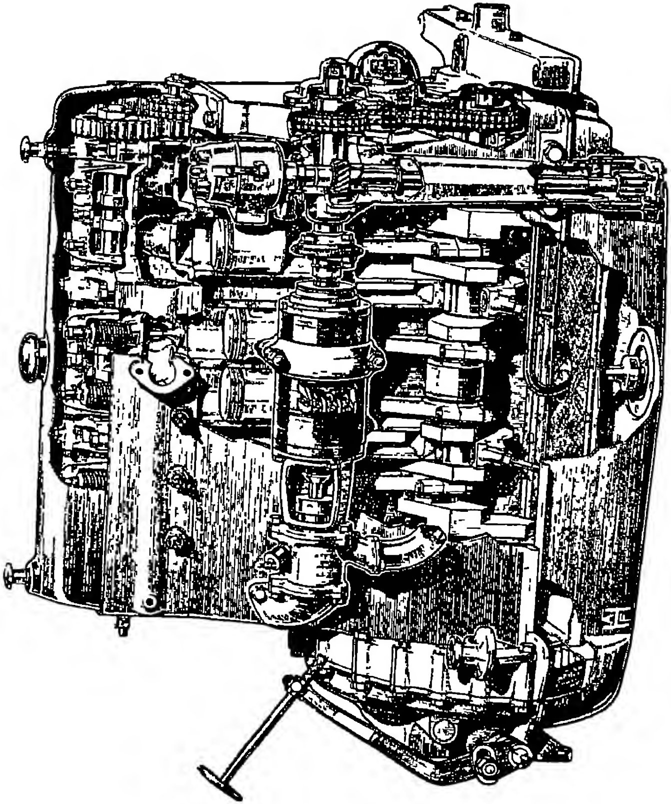


FIG. 5. MORRIS SIX-CYLINDER ENGINE

system. In this engine there is no continuous circulation of oil from the pump; the use of roller bearings instead of sliding or plain bearings makes such a system possible. A slow-acting pump worked by the engine draws oil from a compartment in the tank, and delivers it through the drip sight feed, from which it passes into

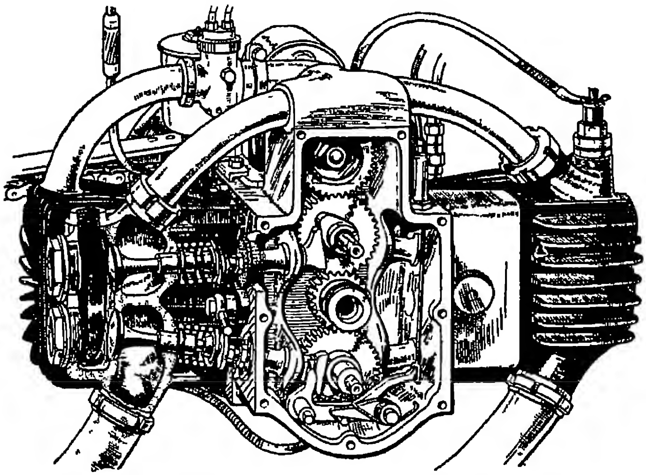


FIG. 6. DOUGLAS TWO-CYLINDER CYCLE ENGINE

the crankcase. The rate of flow is clearly visible, and may be regulated as required. When extra oil is required as at starting, the hand pump may be used, the oil being fed in this case also through the sight feed.

The National single-cylinder stationary engine, Fig. 8, is of heavier design than vehicle engines, and has a heavy flywheel for continuous steady running. All the internal parts are lubricated by splash from dippers on the ends of the connecting rods, the oil in the splash troughs being kept at a constant level by

eding scoops worked by rings running on eccentrics ı the crankshaft.

BEARINGS

Divided or split bearings are used for the main ınkshaft and for the big ends of the connecting rods

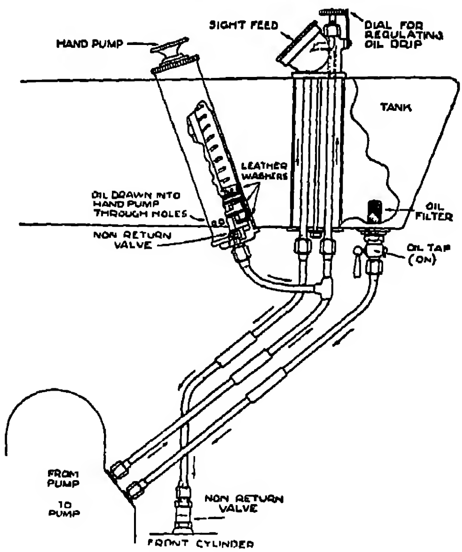


FIG. 7. DOUGLAS DRIP-FEED LUBRICATION SYSTEM

.ll petrol engines, with the exception of a number of or-cycle engines where ball or roller bearings are loyed.

'hite metal or babbit metal is invariably used for g these bearings. There are various reasons for The white metal yields very slightly on the high ts and wears to a smooth mirror-like, nearly friceess, surface If for any reason, such as lubrication re, the bearing runs hot, the white metal melts,

leaving the steel journal uninjured. The white metal lining can be comparatively easily and cheaply renewed, whereas a damaged journal is expensive to

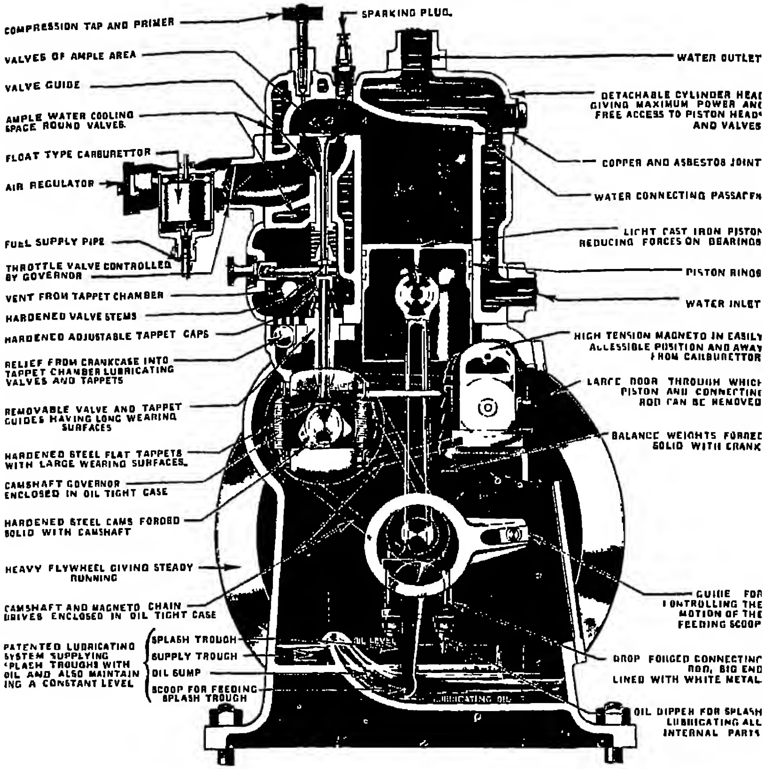


FIG. 8. NATIONAL SINGLE-CYLINDER STATIONARY ENGINE

regrind, and might even necessitate a new crankshaft. A brass or bronze bearing which overheats to the point of seizing could damage the steel journal seriously.

In the modern petrol engine with its possibility of

4000 or more revolutions per minute, the inertia forces at the ends of the up and down strokes may cause momentary forces of over a ton between the big-end lining and the journal, these forces at such high speeds being greater than the forces due to the explosion

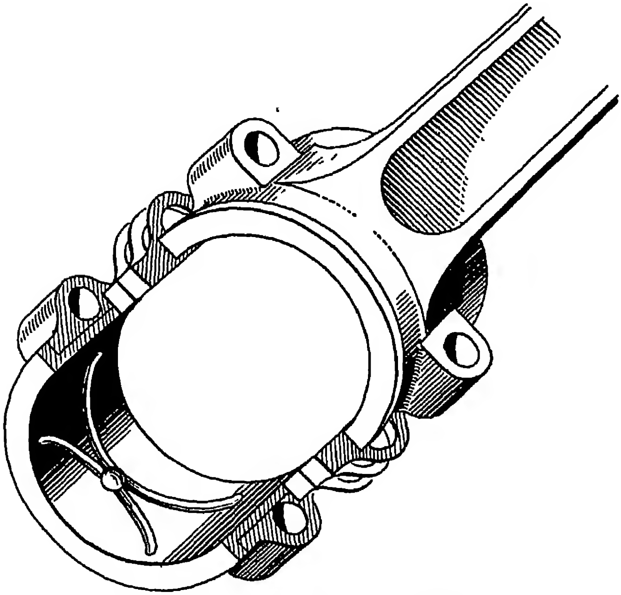


Fig 9. Parts of Big-end Bearing

pressure. Such inertia forces are due to the mass or weight of the piston, connecting rod and gudgeon pin, which have to be brought to rest at the end of each stroke and started on the return stroke.

Crankshaft and big-end journals are necessarily of substantial dimensions in view of the forces which the crankshaft has to resist, and their rubbing speeds are therefore very high, that is, the white metal and steel

are sliding over one another at a great rate. Although the friction is small, much heat is generated owing to the speed and the oil is an important factor in conducting this heat away, so that oilways in the bearing require careful attention.

These considerations make it clear that the greatest care is necessary in fitting bearings, and explain why failures of big-end bearings are more frequent than failures of other parts.

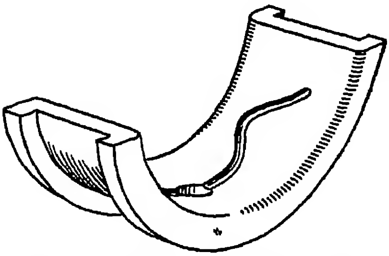


Fig. 10. Flanged Bearing Liner

Big-end and main-journal bearings are very similar, and the practical remarks which follow apply equally to both.

The separated parts of a big-end bearing, shown in Fig. 9, consist of the steel or duralumin connecting-rod end, and the bearing cap, each with a flanged liner of the pattern shown in Fig. 10, also the shims or distance packing pieces.

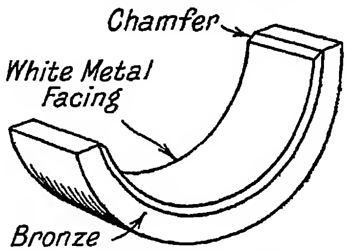


Fig. 11 Crankshaft Bearing Liner

The liners may be solid die-cast white metal, or may consist of bronze shells with a facing of white metal on the inside and on the flanges to form the wearing surfaces. An unflanged bronze shell with a white metal facing is shown in Fig. 11. This liner is not grooved, in accordance with practice sometimes followed in bearings where oil is supplied under high pressure.

A section of the usual inverted main crankshaft bearing is shown in Fig. 12; the liners in this case are of the flanged pattern, and are kept from turning by a projecting pin (*A*) registering with a hole in the cap.

A somewhat lighter big end may be obtained by casting the white metal direct into the end of the rod

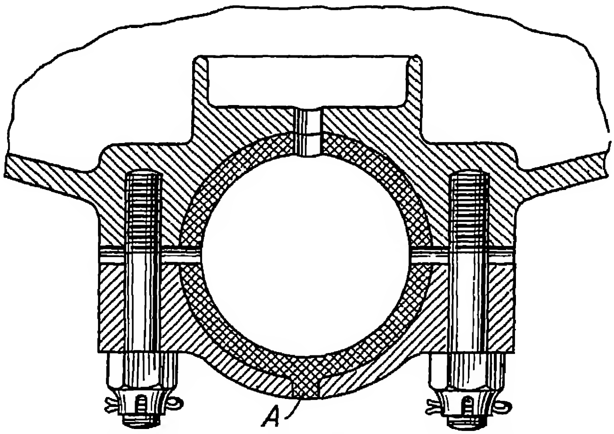


FIG. 12. INVERTED CRANKSHAFT BEARING

and into the cap, the two being provided with undercut or dovetailed recesses to retain the liner firmly and permanently in position

The liners always come from the machine shop or die-caster very nearly correct, and requiring nothing more than hand work In the production shops the accuracy of this previous work may be such as to obviate entirely any fitting. Each liner in such shops is carefully tested before assembly, the tolerances or departures from the specified sizes being less than $\frac{1}{1000}$ in. The shaft journals and the recesses for the liners in the crankcase, big end, or caps are similarly tested for accuracy, with the result that when firmly

bolted up the bearing fits with just the right degree of tightness.

Many manufacturers rely, however, to a greater or less extent, upon hand finishing, and all overhaul or repair work requires careful and skilful hand work.

The meeting surfaces of the two half liners engage

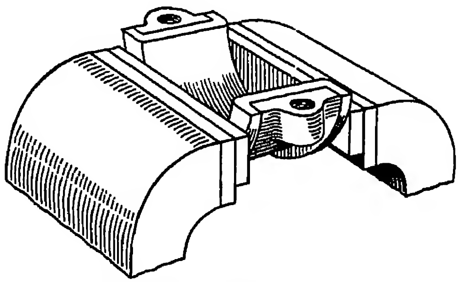


FIG 13 BEARING CAP

over substantially their whole area, and are clamped hard up by the bolts. Care must be taken to obtain a good area in engagement, each liner being held carefully in the vice (Fig. 13), and draw filed where necessary or rubbed on a sheet of emery as shown in Fig. 19. Wood or other soft packing pieces should be interposed between the bearing and the jaws of the vice.

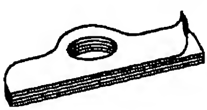


FIG. 14. SHIMS

The use of shims reduces this work considerably. Shims consist of a number of very thin metal sheets or laminae. They are generally of brass and may be separate, but are more frequently stuck or soldered together. Fig. 14 shows a shim pack stuck together, so that one at a time may be peeled off as required by using a pointed tool. When soldered together, the top shim may be carefully and evenly filed off until only the solder of the joint shows.

The fitter or repairer will generally find the question of the best arrangement of oil grooves already settled for him by the makers, but in some cases the cutting of grooves will be left to his discretion. With pressure lubrication systems and narrow bearings, the film of oil is often considered to be most easily obtained by

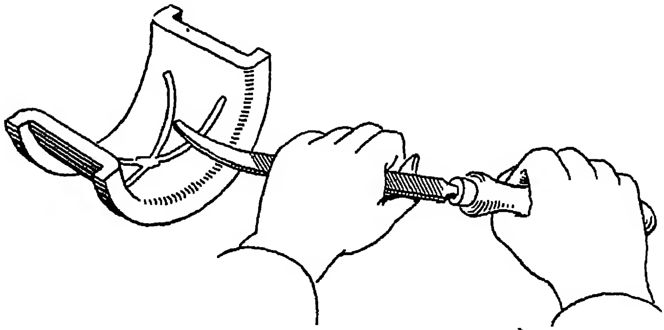


FIG 15 SCRAPING A LINER

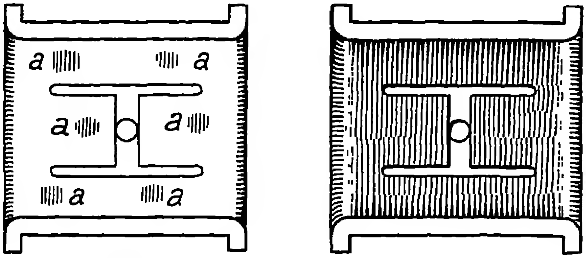
dispensing with grooves, either on both liners or, at least, on the top liners of big-end bearings.

In all cases, the grooves should stop short of the radius at the edge of the bearing. This is particularly important in the case of the main crankshaft bearings in pressure lubrication systems, since the oil which is fed to these bearings under pressure passes through holes in the crankshaft journals to the big-end bearings. If the grooves extend to the edge, that is as far as the radius on the edge, the oil will escape too easily into the crankcase from the ends of the grooves, and the big-end bearings will therefore be starved and will wear and ultimately fail

Fig. 11 shows a liner without grooves. Fig. 10 shows a wavy groove, and Figs. 9 and 15 show the common crossed grooves, an oil-hole being generally provided at

the crossing point. The arrangement shown in Fig 16 is often considered better than the crossed grooves.

Grooves may be cut by the round-nose chisel shown in Fig. 17. The bearing must be firmly but carefully



(A) (B)

FIG 16 SCRAPING IN BEARING LINERS

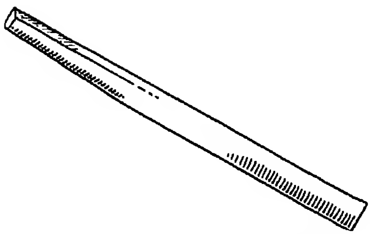


FIG. 17. ROUND-NOSE CHISEL

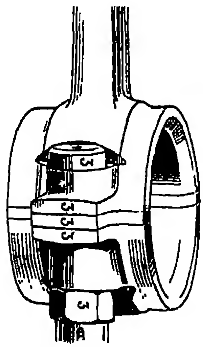


FIG. 18 MARKING BIG-END COMPONENTS

held, and a light hammer used. Burred edges must be scraped down.

All parts of a bearing must be carefully numbered to ensure correct assembly. This is generally done by the makers, and when replacement parts are fitted they should be similarly marked. Fig. 18 shows the method

marking all the components on the big end at the ırd cylınder from the front on a Rover sıx-cylinder ıgıne. Where number stamps are not available, a ntre punch, chisel, or sharp file may be used.

In some cases the bearıngs are only roughly fitted the journals, but are left tight and are "burnt in." ıe engine is run without oıl, or the crankshaft is iven from some external source of power, until the

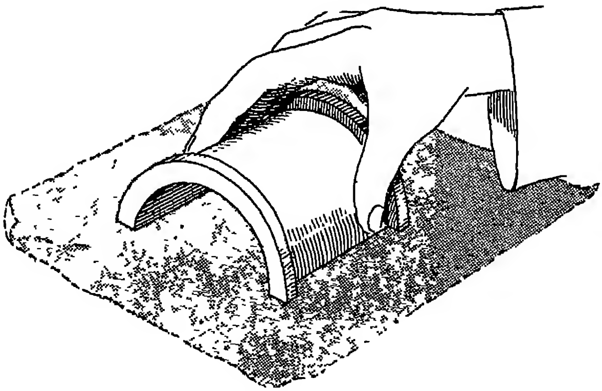


FIG. 19. RUBBING DOWN ON EMERY CLOTH

ırings heat. They are then allowed to cool down. e running operation is repeated untıl the working faces are rubbed level and brıght, the tightness of bearing being varied as requıred by removing shims, by the use of the file on the abuttıng edges.

Ihe best results are obtaıned by hand scrapıng the ɜrs. The crankshaft journal is cleaned and smeared h Prussian blue (from an artısts' supplıer), or wıth lead or lampblack mixed with oıl The bearing, er beıng completely assembled and bolted up firmly, vorked to and fro several times When taken apart, high poınts on the lıners will be clearly marked, as

shown by way of example at (*A*) *a* in Fig. 16 The marked parts must be removed by the scraper, as illustrated in Fig. 15. These operations of fitting to the journal and scraping must be repeated until only small gaps are left between bearing points, as shown at (*B*) in Fig. 16 At the same time, and this is of equal importance, shims must be removed, or the abutting edges of the liners must be filed or scraped to give just a sufficiently tight fit on the journal when bolted up hard. Never use emery cloth or powder or any abrasive on the working surface, as the bearing or journal may be ruined thereby. The abutting edges may, however, be rubbed down on a sheet of emery cloth resting on a flat surface, this method being shown in Fig. 19. In the case of a connecting rod, this fit may be tested as shown in Fig. 20. The bearing, while holding to some extent, should allow the connecting rod to swing over under its own weight.

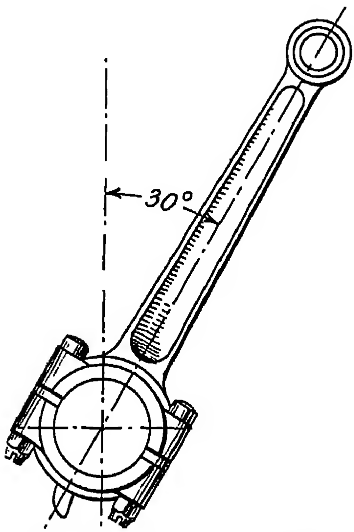


FIG 20. TESTING FIT OF BIG END BEARING

Fig. 21 shows a suitable scraper made from an old file, the rounded end being ground and preferably finished on an oilstone.

PISTONS

At the present day, cylinders are invariably made of

cast iron, while pistons are made either of cast iron or of an aluminium alloy. Cast iron pistons are most generally satisfactory, and present less difficulty in fitting. Aluminium, on the other hand, is lighter, and has therefore advantages over cast iron for high speed engines, although it presents difficulties of design and workmanship, and does not wear as well as cast iron.

A specially light pattern of cast iron piston is shown

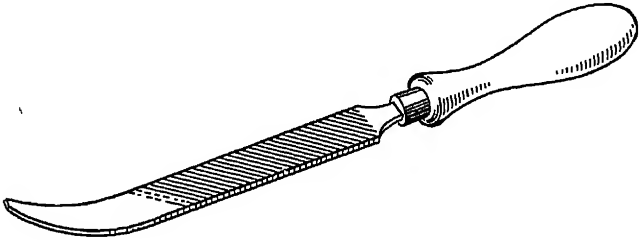


FIG. 21. STEEL SCRAPER

in Fig. 22. The internal ribs give strength to the crown and walls, and allow the walls to be cast very thin. This piston is made by Messrs. Laystall.

Aluminium alloy pistons are of greatly varying design, as they are still in the development stage. Many designs include saw cuts to form an insulating air gap separating the heated crown from the skirt. An example, made by the Light Production Company, is shown in Figs. 23 and 24

The crown of the piston is exposed to the full pressure and very high or flame temperature of the hot gases. Under working conditions, it expands more than the cylinder walls and more than the skirt. When both cylinder and piston are cold, the clearance of the skirt in a cast iron piston, that is the difference in diameter between piston and cylinder, should be between ·00075

and ·001 in. per inch of diameter. Thus a piston 3 in. in diameter should have not more than ·003 in. clearance A three-thousandth feeler gauge should only just enter between the skirt and the cylinder wall. The upper part of the piston, owing to its greater clearance, is given substantially more clearance (see

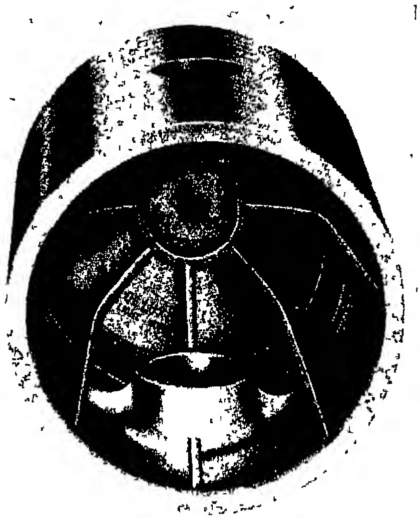
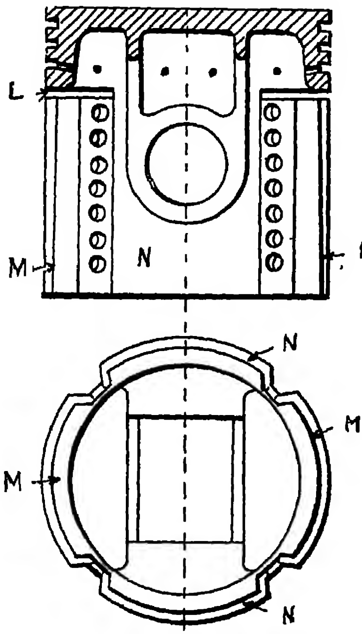


FIG 22 LAYSTALL CAST-IRON PISTON

Fig. 25, in which this effect is greatly exaggerated). The clearance at the head may amount to ·003 or even more per inch of diameter When these clearances are substantially exceeded as the result, for instance of wear, the pistons may knock as they move from side to side, particularly when they are cold at starting. When the clearance attains about ·0025 in. per inch of diameter, the cylinders should be reground and new oversize pistons fitted.

The clearances mentioned above for cast iron pistons are safe for all conditions, but some manufacturers do not follow them exactly Hard and fast rules cannot be laid down, as different designs of engines differ substantially In general, smaller clearances should not be used unless the pistons are fitted very carefully to avoid high spots To do this, insert the piston in the cylinder and work it well round. Then remove and take off the high spots, which will show bright where it has been binding , use a fine file and finish with fine emery cloth When the pistons can be pushed through without binding, they should be connected up to the crankshaft, and the engine run slowly for two hours, using plenty of oil In the case of motor vehicle engines, an engine speed equivalent to 20 miles per hour on top gear should not be exceeded for at least 200 miles.



FIGS 23 AND 24 ALUMINIUM PISTON

Clearances in the case of aluminium alloy pistons vary greatly with the design, and the manufacturer's advice should be followed in each case. All aluminium alloys expand much more rapidly with heat than cast iron, and the clearances when cold must therefore be

substantially greater. For pistons with solid skirts, ·0016 in. per inch of diameter is a fair average value The clearance at the top of the piston is, of course, greater. When the skirt is split the clearance may be substantially less, as yielding then takes place during expansion, and the piston cannot readily bind in the cylinder.

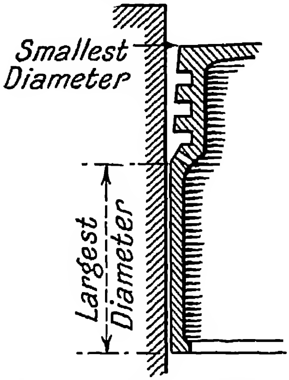


FIG. 25 PISTON CLEARANCE

There are many designs of aluminium alloy pistons intended to overcome the expansion difficulty, that shown in Figs. 23 and 24 being one example. The two faces *M* which take the side thrust are separated from the hot crown by saw cuts *L*; the side faces *N* are joined on as usual to the upper part, while a series of small holes partly insulates the parts *M* and *N*. The piston changes its shape slightly as the temperature rises just sufficiently to compensate for the higher rate of expansion of the aluminium alloy of the piston over the cast iron of the cylinder.

PISTON RINGS

Piston rings are fitted to a piston for two main purposes. They are intended to keep the hot gases and any unburnt fuel from passing downwards, and must also prevent lubricating oil from passing upwards while permitting proper lubrication of the cylinder walls and pistons.

Piston rings are invariably made of cast iron, the best rings being made from centrifugal castings; that is, the annular castings from which they are turned

1 parted off are cast in rapidly rotated moulds, so it centrifugal force is utilized to obtain a close firm ting.

The rings must fit the cylinder walls and the sides the groove closely, l they must be htly shallower than grooves, so that the on cannot force m sideways against cylinder walls.

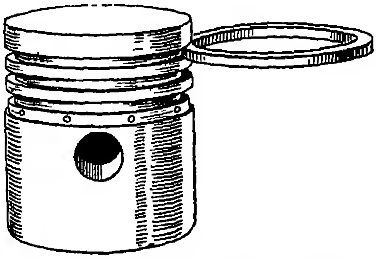
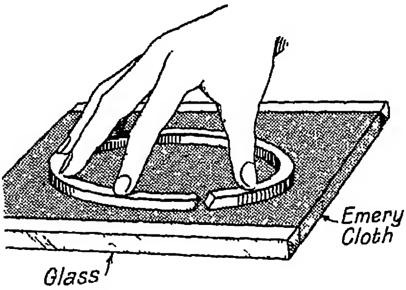


FIG 26. TRYING PISTON RINGS

'he grooves in the on must be cleanly ied with sharp cor-, and the rings fully fitted to the grooves The method of trying one in the other is shown in Fig. 26 The ring t fit without slack, but quite freely at all points when rolled round the groove. Manufacturers now cut the piston grooves and grind the faces and edges of the rings to gauge, so that they fit without hand work; but in repair shop individual fitting of each ring to its ve is often necessary. The edges of the rings may used and finished by rubbing them gently on a of fine emery cloth (Fig 27), stretched over a plate or a sheet of plate glass or glued to a flat l.



IG 27 RUBBING ON EMERY CLOTH

Piston rings are to-day practically always of the concentric type, that is, of uniform depth all round. When the groove is cut and they are sprung into the cylinder, they should at first be left slightly overlapping. The lower ring in Fig. 28 illustrates this. The ring will in all probability be found to bear hard close to each side of the gap and on the opposite side, and to be clear of the walls at intermediate points. The high points may be carefully removed with a fairly fine file until the ring is bearing more or less all round; the gap may then be enlarged until it is ·006 in. when the ends are in line, as shown in the upper part of Fig. 28. This gap allows the ring, which becomes hot, to expand more than the cylinder walls without binding. One method of holding the ring in the vice while filing the ends is shown in Fig. 29, a wood strut *A* being inserted to spread the ends apart.

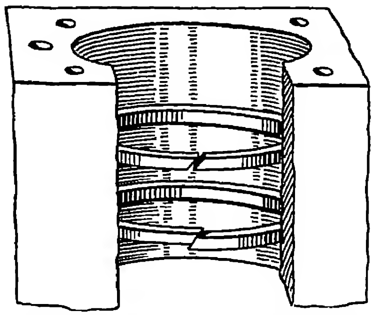


FIG. 28. FITTING PISTON RINGS

If the filing of the face has been carefully done, the ring is often left to run itself in when the engine is put into use, but a better finish and fit may be obtained by lapping. In this process the ring is worked to and fro in the cylinder, using an abrasive mixture of carborundum and oil or crocus powder or valve grinding

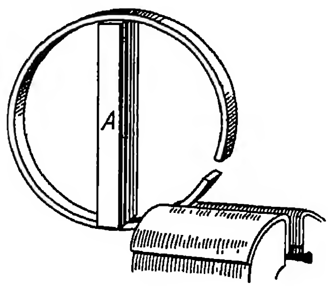


FIG. 29

compound, the abrasives used being the finest obtainable. Emery powder is not suitable for this work, as the particles tend to embed themselves in the metal and cause undue wear subsequently.

During the lapping process, the ring may be held in an old piston, or secured to the end of a cylindrical block of wood specially turned for the purpose, as shown in Fig. 30. The ring is held on a reduced part

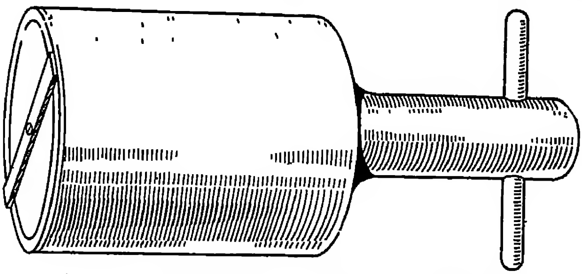


FIG. 30 PISTON RING HOLDER FOR LAPPING

at the end by a light cross-bar secured by a wood screw. In this, as in all other cases where an abrasive is used, the greatest care must be taken to remove all traces by washing carefully with paraffin or petrol.

Considerable care is necessary when removing rings from or replacing them in the grooves, as they are easily broken or may be distorted so as to render them useless. If any difficulty is experienced, the use of three strips of thin metal is recommended (Fig. 31). If two of these are inserted between the ring and the piston, one on each side of the gap, and the third is similarly used on the opposite diameter, the ring can easily be slid on or off or moved over an intervening groove.

A certain amount of oil is thrown on to the cylinder walls by the connecting rods and is necessary for lubrication, but too much may pass above the piston and cause

trouble. Various constructions and arrangements of rings are adopted to scrape the surplus oil off the cylinder walls. In Fig. 23, it will be noticed that the lowest of the three ring grooves is connected with the interior of the piston by a number of small holes. The surplus oil scraped off the walls on the down stroke is discharged inwards through these holes. A similar, but somewhat more effective, arrangement is shown in Figs. 25 and 33. A special scraper groove and ring is sometimes provided at the lower edge of the piston skirt.

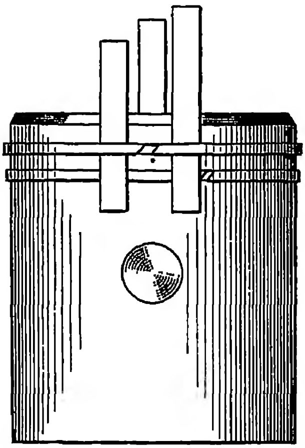


Fig. 31. Ring Removal

A scraper ring is most effective when it has a narrow periphery or face engaging the cylinder wall with a sharp edge for scraping downwards The Rover scraper ring (Fig. 32) is tapered with this object, and must always be placed in its groove with the narrow side upwards. This can be verified by the two centre dots (indicated by arrows) on the top face of the ring Several effective patterns of scraper rings are shown in Fig. 33, *A*, *B*, and *C*, that shown at *B* having a circumferential recess and slots passing right through to the inside. Ring *A* should always

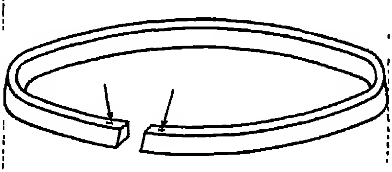


Fig. 32 Scraper Ring

be arranged as shown with the chamfered edge upwards.

GUDGEON PINS

Piston pins or gudgeon pins may be fixed in the piston or in the small end of the connecting rod, or

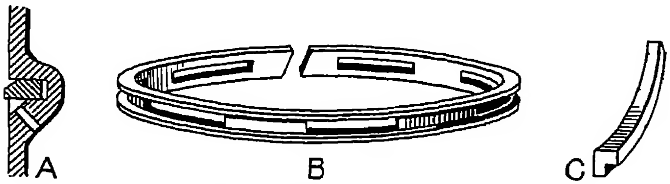


FIG 33. SCRAPER RINGS

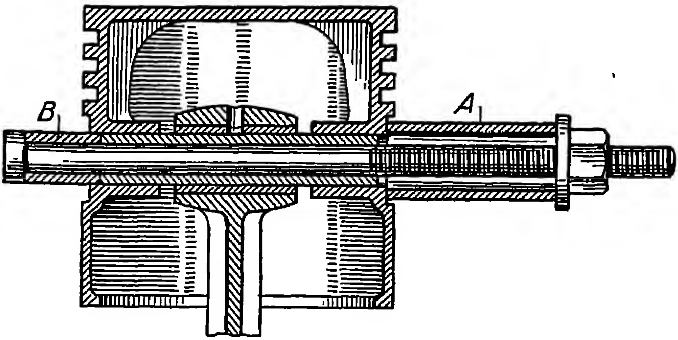


FIG 34. REMOVING GUDGEON PIN

may float or work in either. Case-hardened steel tubes are invariably used.

When the gudgeon pin is fixed in the piston it is often made a light push fit. Two methods of forcing the pin in or out are shown in Figs. 34 and 35. In the former a long bolt and large and small tubes *A* and *B* of various lengths may be used as required. In the method shown in Fig. 35, a bolt *a* screwed through a

plate *b* held by metal strips *c* pushes the pin into or out of the piston. A vice with widely opening jaws may be used in a somewhat similar fashion, but the piston must not be gripped directly in the steel jaws. It is safest to use lead clamps, although sheet brass, copper or zinc clamps may serve if care be taken.

While the gudgeon pins must not easily move when once pushed in, they must not be tight enough to require any substantial force to insert them. The correct fit is known technically as a push-fit, and is such that the pin can be pushed into the hole, but will not be free enough to turn without seizing. A push-fit for pins and holes of the diameter in question may be expressed in figures as follows: the pin should be smaller than the hole by at least ·00025 in., and not more than ·00075. These figures are termed the "allowances" for a push-fit.

A fixed pin may be made a somewhat easier fit—a good running fit—and held by a set-screw entering one of the bosses in the piston (Fig. 36). The tapered end of the set-screw should jam in a corresponding hole in the pin, and the set-screw should be locked by a split pin or cotter pin.

Should a gudgeon pin for any reason work loose, and move endwise, the end would in all probability score the wall of the cylinder and do considerable damage. To prevent this possibility, plugs of soft metal, such as copper, brass, or aluminium, are often inserted into the ends of the gudgeon pin holes. They are tapped lightly to jam them into position, as shown at *A*, Fig. 36, and only make contact with the cylinder walls in an emergency.

Somewhat thicker plugs may be used similarly to prevent end movement of floating gudgeon pins.

Fig. 37 shows a method of fixing the gudgeon pin in the small end of the connecting rod, bronze bushes

being fitted in the piston. The clamping screw engages a slot in the pin and clamps the split end of the connecting rod on to the gudgeon pin, which is further held in position by the screw engaging a groove on one side. The screw must be tightened by a box spanner, and may be locked by a spring washer and split pin or wire. The Morris gudgeon pin, Fig. 38, is fitted in this way.

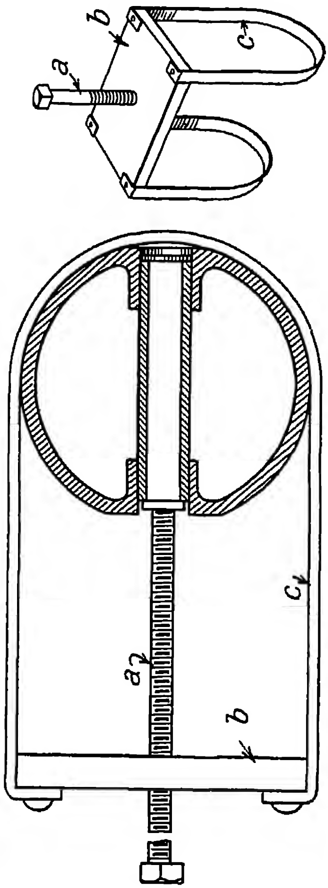


Fig. 35. Removing Gudgeon Pin—Another Method

Gudgeon pins work in bronze bushes in the end of the connecting rod or in the bosses of the piston. Although the forces transmitted are considerable, there is very little rubbing movement and correspondingly little wear. The wear which does take place is largely in the bush.

The most convenient method of inserting and removing gudgeon pin bushes requires a vice with sufficient opening (Fig. 39), and short lengths of tubing of suitable dimensions. The smaller piece of tube must be a little smaller externally than the exterior of the bush, while

the larger tube must fit over the bush easily. An alternative method of removing or inserting the gudgeon pin bushes requires a bolt, washer, and suitable short length of tube (Fig. 40).

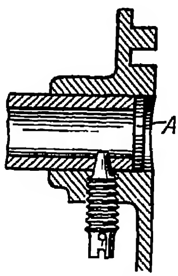


Fig. 36 Securing Gudgeon Pin in Piston

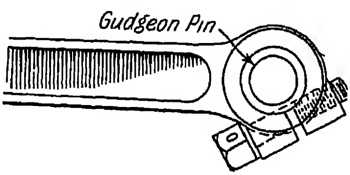


Fig. 37. Securing Gudgeon Pin in Connecting Rod

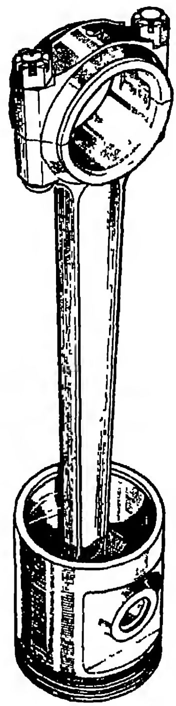
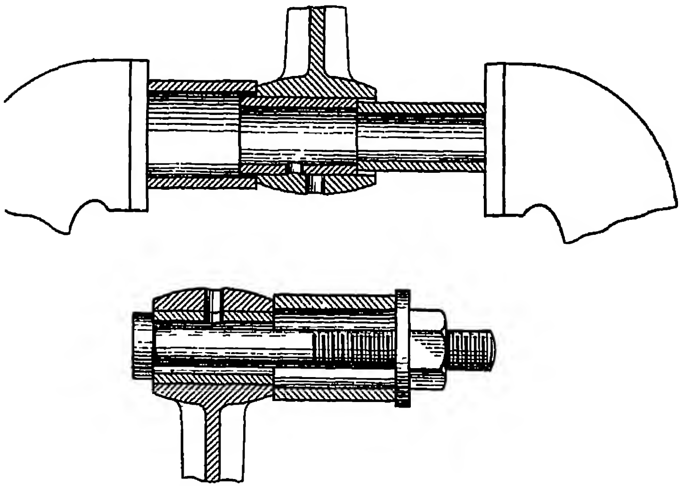


Fig. 38. Piston and Connecting Rod

The bushes are turned a driving-fit, so as to be gripped firmly when pressed in. They should be bored a little larger than the pin to allow for the closing in when they are forced into the rod. When new or after renewal of bushes, the pin and bush should hold slightly,

ı that when the pıston and connectıng rod are held ɜrtically the connecting rod should not move, but ɩould fall under its own weıght when pushed 10 or 5 degrees from the vertical. Fıg. 38, which shows a orris piston and connecting rod, illustrates this point. ıch a fit may be obtained by reamering out the hole



FIGS 39 AND 40. TWO WAYS OF REMOVING GUDGEON PIN BUSH

by driving the gudgeon pin in (when the two are arly right), so as to burnish the surface of the bronze. ıe pin may be eased in when applying this last ɜthod by a scraper if too much force appears to be cessary.

A small amount of wear can be taken up and a good obtained by removıng the bush, tinning the outside solder and forcing ıt in again This will close the sh ın slightly on to the pin. Care must be taken at the tinnıng is uniform, and that the bush is

replaced in exactly the same position as before. To ensure this, the end of the bush and an adjacent point on the rod should be marked, as for instance, by a centre punch.

PISTON AND CONNECTING ROD ASSEMBLY

The pistons and connecting rods must be connected finally to one another, and the big-end bearings ready for assembling finally to the crank pins before the pistons are inserted into the

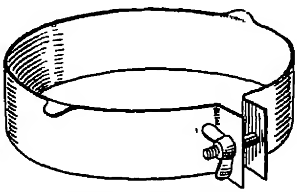


FIG. 41. PISTON RING COMPRESSOR

cylinders. It is preferable to insert them from the lower ends of the cylinders, since there is generally a good slow chamfer which closes in the rings as the piston is pushed in. When the piston is made without a chamfer either at top or bottom, the rings must be closed before they will enter the cylinder. No force must be used to drive the piston in; the rings should be so well closed that the piston will slip in without effort. Each ring may be closed in its turn by a piece of string which will slip off as the ring enters the cylinder. The rings may be closed in by the clamping device shown in Fig. 41. The bent ends and the

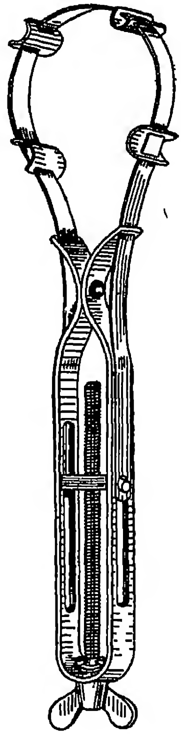


FIG. 42 ANOTHER RING COMPRESSOR.

tongues on the edge prevent the clamp from being drawn into the cylinder as the piston is pushed in.

A piston ring compressor adaptable for any usual size of piston is to be preferred for general use. The "Beacon" shown in Fig. 42, is of this type, adjustment being effected solely by a wing nut.

The piston and rings before assembly in the cylinder must be thoroughly cleaned by washing with petrol and then well oiled, since oil will not work up to the upper rings and upper part of the piston until the engine has run some time.

Care must be taken that the pistons are not pushed through too far as the top ring may spring out, and in some engines would be very difficult to put back.

The gaps in the case when three rings are used should be spaced out equally round the piston before assembly. When two rings are used, the gaps should be diametrically opposite.

JOINT-MAKING

Methods of making the various gas, water, or oil-tight joints found on petrol engines vary according to the purpose of the joint, and the condition of the metal surfaces.

Where the surfaces are truly flat and well finished, a coating of gold-size, with or without brown paper, may be all that is necessary; but where the surfaces are very rough or out of truth, a thick jointing material such as cork may be essential. Instances requiring the use of cork occur in roughly-finished sumps of cast iron or aluminium, or in pressed steel sumps where the flange is thin and easily deformed by the securing studs or bolts.

Where the cylinder casting is separate from and is bolted down to the upper part of the crank chamber, the metal surfaces must be a good fit to avoid any

strain or distortion of the parts. If this result has not already been achieved by accurate machining, the surfaces must be filed or shaped to fit, the high places being indicated by the usual mixture of red lead or lamp-black with oil. A rough file should not be used. This is, of course, a comparatively rough job not comparable with the fitting of a bearing to its journal, the object being merely to obtain a sufficient number of bearing points so well distributed as to avoid strain. This joint has merely to be oil-tight, the oil not being under pressure, so that gold-size (with or without paper) or other jointing compound is sufficient. In many engines the cylinders and the upper part of the crank case are made in one, so that this joint is eliminated.

Nearly all modern petrol engines are made with detachable cylinder heads The joint must be proof against leakage of the hot gases, which are at a very high pressure, and also the cooling water. The water is not under pressure, but is much more penetrating than air or gas.

Fig. 43 shows the Morris detachable head and the jointing gasket lifted, the valves, ports, and the water circulating passages being visible. In view of the high temperatures to which the joint is subjected, asbestos is essential as the basis for the copper-asbestos gasket.

When lifting the head, care must be taken to avoid damaging the gasket or the edges of the metal faces. A screwdriver or chisel should not be driven in to break the joint. The head if stuck down firmly can be loosened by rapping it in several places with a raw-hide hammer or wooden mallet, or a steel hammer with wood interposed. If the compression is good, the engine may be turned sharply by hand, with the switch "off," when the compression pressure will probably break the joint.

When the head is loosened, it may readily be removed

if set bolts have been used for bolting it down When studs are used, greater difficulty is often experienced, and care must be taken to lift it evenly without tilting.

Projections are sometimes formed on the head near the joint to enable a large screwdriver or a tyre lever to be used to lever the head up without damaging the gasket. The head may also be lifted by standing over

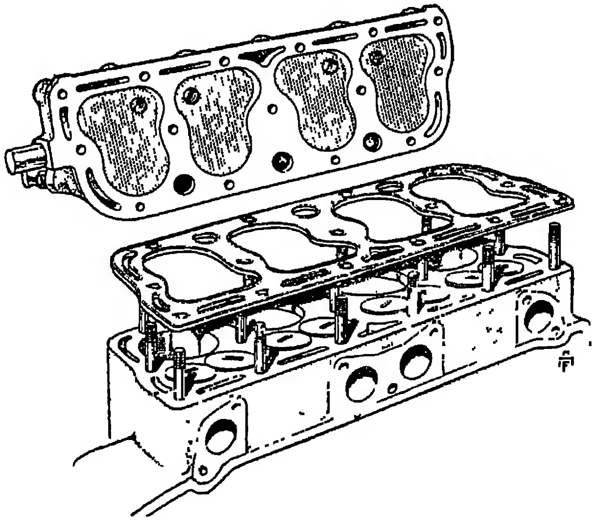


FIG. 43 DETACHABLE CYLINDER HEAD AND GASKET

the engine and grasping the water connection or other fittings. Attachments such as Tee-headed tools screwed into the holes from which the sparking plugs have been removed may also be used for this purpose.

Special tools are sometimes provided, an example being shown in Fig. 44 in connection with an Austin engine. Screwed spindles with tommy bars are inserted in place of the two end sparking plugs, until their ends find a firm bearing on the top of the cylinder casting.

They are then turned together so as to lift the head uniformly.

Before placing the detachable head in position, the surfaces must be carefully scraped and cleaned. In the case of new work, the finish should be such that no fitting is necessary, but instances occur, in connection with both new and old work, where some attention

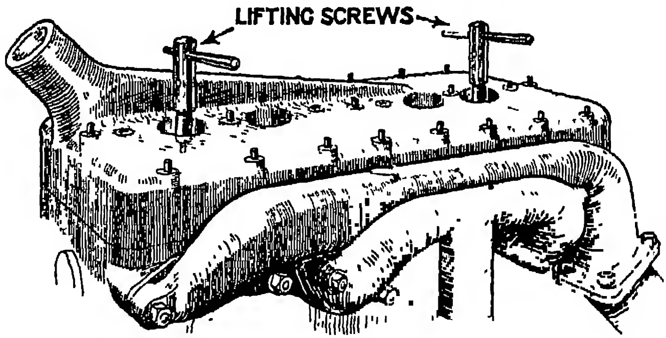


FIG 44. LIFTING DETACHABLE CYLINDER HEAD

may be required. A scraper should be used in preference to a file, and attention should be given particularly to the narrow parts between cylinders and round the water passages.

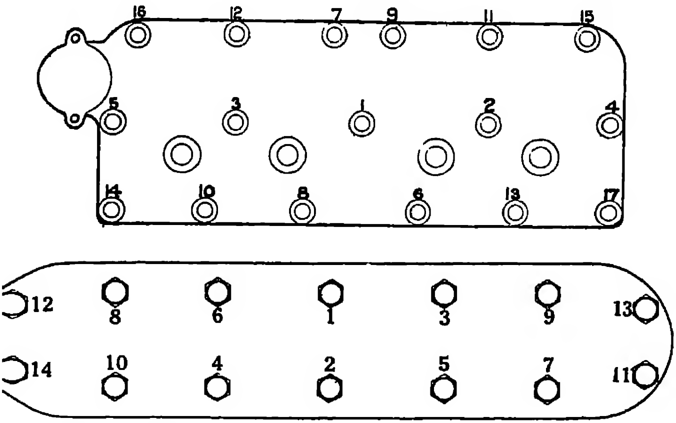
An old gasket may be used again if it has not been damaged in any way. Care should be taken when a new gasket is employed that there are no burrs round the stud holes, and that the gasket clears the cylinder bores.

It is advisable to coat the set-screws or studs lightly with a graphite grease to prevent rusting or corrosion, and to ensure ready removal when necessary at a later date.

Manufacturers differ in their advice as to the use of a dressing on the gasket. In some cases, when the metal

aces are particularly well finished, nothing may be ıeeded, or a lıttle grease on both sides may be suffici- ›nt. In general, both faces of the gasket should be ;iven a thin even coat of gold-sıze or boiled linseed oil ; ›r the special jointing compounds made for this purpose nay be used.

When the head is in position, the nuts or studs



FIGS. 45 AND 46. ORDER OF TIGHTENING CYLINDER HEAD STUDS

ıould all be screwed on finger tıght, and should then e fastened down each a little at a time and ın such a ɘquence that the head is tightened down unıformly. 'ypical examples of the best sequence to follow ıs ıown in Fıg. 46 in connection with a Trıumph ›ur-cylinder engıne, and in Fig. 46 in connection with Wolseley sıx-cylinder engıne. In each case a start made at or near the centre, and the nuts are there- fter tıghtened oppositely in pairs working away from ıe centre. Each nut should only be tightened a lıttle t a time, say half a turn at first, so that all the nuts

will be gone over several times, the order given being followed on each round After the engine has been warmed up by running a short time, it will be found that the nuts can be tightened up a little more

The joints between the exhaust manifold and the cylinder casting or cylinder head must be made with asbestos. This applies also to any other exhaust joints. Copper-asbestos gaskets or compressed asbestos as supplied in sheets is preferable to soft asbestos sheets. Gold-size or shellac dissolved in methylated spirit or the special heat-proof jointings may be used.

For the inlet manifold and for all carburettor joints, asbestos may be used, but the softer materials enable tight joints to be made more easily without any dressing; some of these materials have a rubber basis and will not stand heat.

For other oil and water joints, such as those on the oil pump, water pump, and oil filter, thick brown paper dressed on both sides with gold-size is quite reliable.

VALVES

Conical lift valves or poppet valves in the case of four-stroke engines are generally used, and may be arranged at the side opening upwards, or overhead opening downwards The necessity for careful fitting and adjustment will be apparent when the strenuous conditions under which they work are appreciated Both the inlet and the exhaust valves when seated have to withstand the flame heat and high pressure of the burning mixture, and the exhaust valve is opened in opposition to what is still, after expansion of the hot gases, a considerable pressure. Each valve is lifted rapidly by a cam and lowered by a spring 1000 times a minute in the case of a motor-car engine fitted to a car doing about 30 miles per hour.

A side valve complete with its operating mechanism

as used on Morris four-cylinder engines is shown in Fig. 47.

An example of an overhead valve operated through a push rod is that fitted to Star engines (Fig. 48).

The Morris six-cylinder engine (Fig. 49) employs an overhead camshaft, and each valve is operated by an

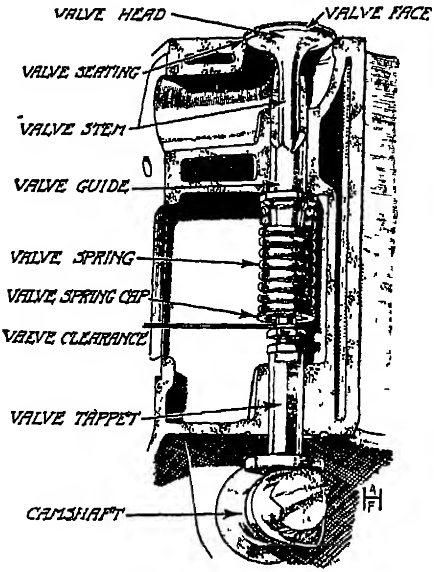


Fig. 47. Morris Side Valve Gear

angle lever or rocker, all the rockers being journalled on a fixed shaft running lengthwise of the engine and above the camshaft.

The primary functions of a valve are twofold · (1) It must close and seal the valve port against escape of the mixture, and (2) it must, when required, lift rapidly, give a clear and unobstructed passage for the flow of incoming mixture or the escape of burnt gases,

and then close sharply. The stem must fit its guide, particularly in the case of the inlet valve, since air drawn in through a leaky valve guide upsets carburation and the running of the engine.

Modern valves are generally made solid, and of 3 per cent nickel steel or other hard heat-resisting

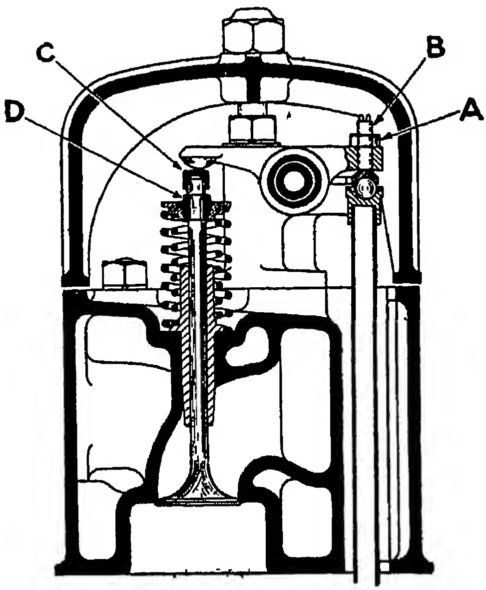


FIG. 48 STAR OVERHEAD VALVE GEAR WITH PUSH RODS

steel alloy, such as tungsten, nickel-chrome, or chrome-vanadium steel. At *A* and *C* in Fig. 50 are shown two examples of valves as they fit on their seats when new, while at *B* and *D* are shown, in a somewhat exaggerated manner, the results of repeated grindings. The valve at *B* is said to be pocketed. In either case, the valve will not be likely to seat properly and tightly, and when

open will give a reduced and obstructed passage for the mixture and hot gases.

When this stage of wear has been reached, the conical surfaces on the valves require refacing or the valve seats require reseating, or both may be necessary. In some cases new valves may be required. If new valve stem guides are necessary owing to wear, reseating of

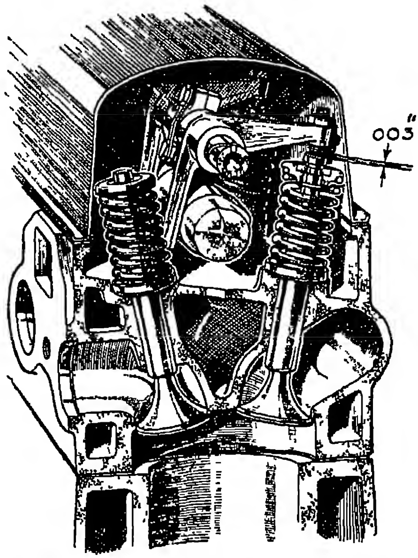


FIG. 49 MORRIS VALVE GEAR WITH OVERHEAD CAMSHAFT

the valve faces should be done after the new guides are in place.

The pocketed valve seat shown at *B*, Fig. 50, may be refaced conically, in which case it will be rather wide, or the surplus material may be removed as shown at *E*. A number of tools is available for this purpose, of which a selection is illustrated.

The J. and S. combined reseater and valve refacer is made in several sizes, and when used for reseating (Fig. 51) may be fitted with stems of any diameter to suit the valve guides. It must be forced down and rotated carefully and steadily by hand. The internal cutter is used for refacing valves, as shown in Fig. 52.

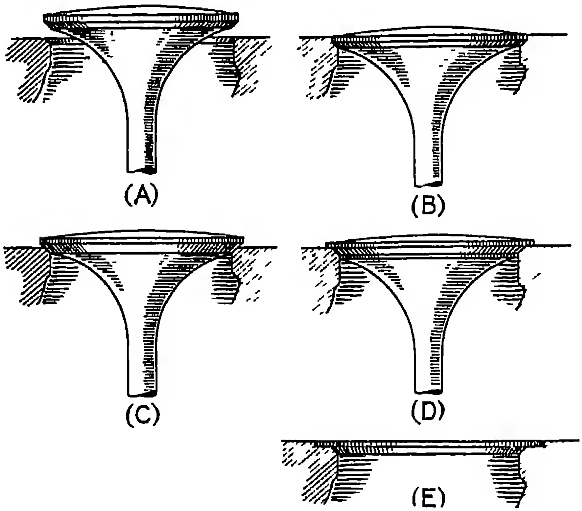


Fig. 50 Valve Wear

The Universal reseating tool (Figs. 53 and 54) made by Mann, Egerton & Co., consists of several conical and flat steel cutters and a handle, adapted to be screwed into the one required for use. The double-coned stem, made in two parts, screwing one into the other fits on to the ends of the valve stem guide. The plain stem projects up through the valve seat, and the hollow handle fits on it so that the cutter is kept central with the guide. The cutters are ground flat on the sides to allow for insertion through small valve pockets.

[t is better to obtain a new valve than to spend much ıe refacing a badly worn or distorted valve.

Cungsten valves which have specially hard faces ıy require grinding, and attachments or brackets are ıd by means of which the valve stem may be held, 1 the valve rotated against an emery wheel driven hand or power.

/alves may be skimmed up in a lathe, care being

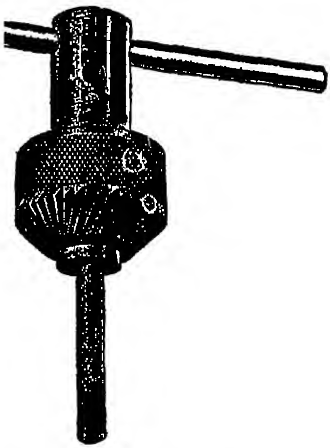


Fig. 51. J and S Valve Reseating Tool

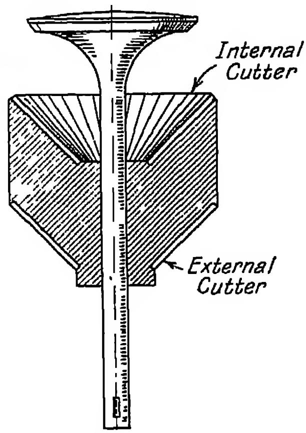


Fig. 52. Refacing Valves

ən that the stem runs true, and that the tool rest ɜt at the correct facing angle.

/hen the valve seats and valves have been accur- y made and finished, little or no grinding of the two ɔther may be necessary. The best results are, how- ·, obtained by grinding for new work, and it is, eover, a normal and fairly frequent repair opera- if the valves are to be kept in good condition. seats and face both become pitted in use as a lt of the severe heat conditions, the exhaust valves

being most affected owing to the flame rushing past the face during the early part of the exhaust stroke.

The best gas seal is obtained with a narrow bearing surface, a width of between $\frac{1}{16}$ and $\frac{3}{32}$ in. being most satisfactory. The narrow surface beds down closer than a wide surface, and remains longer in good condition.

Valve grinding paste or compound consists of carborundum powder with an oily bonding medium, and

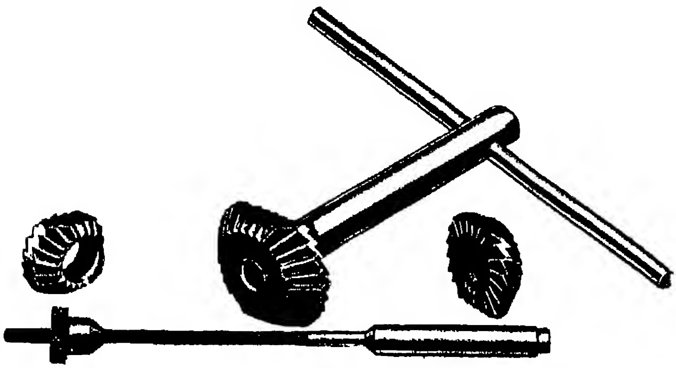


FIG. 53. "UNIVERSAL" VALVE RESEATING TOOLS

is obtainable in coarse and fine grades ; it is often not necessary to use the coarse grade compound. The finer grades, even if they may not cut so quickly, give better final results, and should in every case be used for finishing.

The valve, the seat, and the valve stem guide should be carefully wiped clean with a rag soaked in paraffin, and the faces of the valve should be given a touch of the grinding compound at intervals.

The valve is then put into position, and twisted backwards and forwards in its seating. It should not be pressed too hard, only a moderate force being

ıired. A screwdriver engaging a slot in the head he valve may be used, but sometimes a special tool equired, as in some of the Morris engines where ı valve has two holes. A brace or other tool giving

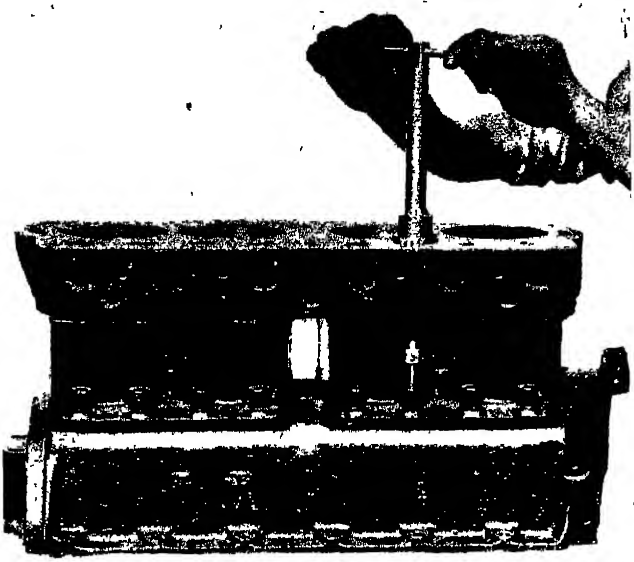


FIG. 54. VALVE RESEATING

tinuous rotation is not good, as it is liable to cause oves.

Ve now come to the secret of good valve grinding. valve should be lifted clear of its seat after every twists. This prevents the compound form working circular lines and cutting fine circular grooves,

which would make it impossible to obtain a good gas-tight fit.

To lift the valve in the most convenient way, a light helical spring is fitted under the head as shown in Fig. 55, which illustrates a side-valve engine with the cylinder head removed The spring is compressed into the port when the valve is pressed down, and raises it when the pressure is relaxed. The compound then

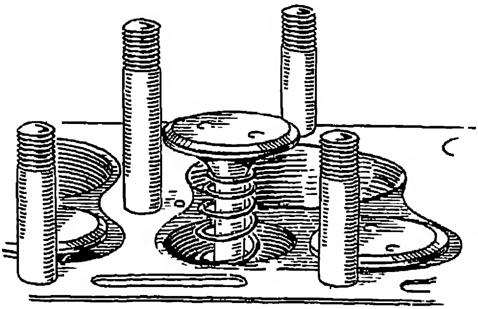


FIG. 55. VALVE GRINDING

spreads out across the face, and the formation of lines or grooves is avoided. The compound may also at times be spread across the face by the finger.

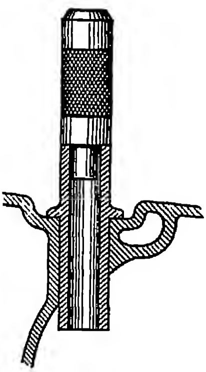
With a new valve, or one that has not been neglected, a few minutes' work should produce an even smooth matt-surfaced ring, running the whole way round both valve seat and valve face. The grinding must remove all signs of pitting, that is, all small black specks or hollows. If the pitting is very deep, the valve must be trued up by a cutter or by grinding as above described, but this should not be necessary unless the valves have been allowed to go for long periods without attention, or unless repeated grindings have brought about the conditions illustrated in Fig. 50.

When the regrinding has been completed every

ticle of abrasive must be removed, as any particles in the engine and getting into the working parts ;ht cause serious trouble. Rag soaked in paraffin, nentioned previously, should be quite effective.

VALVE GUIDES

n some engines, particularly motor vehicle engines \merican origin, the valve stems work direct in the



56 INSERTING VALVE STEM GUIDE

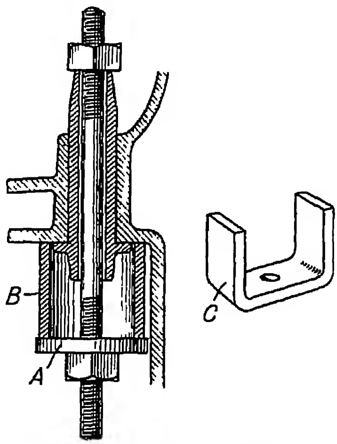


FIG. 57 REMOVING VALVE STEM GUIDE

nder or cylinder head casting, but in nearly all ish designs, detachable valve stem guides are pro- d. These are made of bronze, and are generally sed into position. With all new work, the parts ıld come from the machines to the fitter ready for mbly, but the fitter may have to verify that the e stem guide and the valve seat are true with one her.

ery little wear on either the valve stem or the guide for attention, as mentioned previously, in view of

gas leakage. Where there is no separate valve guide, the hole should be reamered out $\frac{1}{64}$ in. oversize, and a valve with a corresponding oversize stem fitted. If excessive wear has taken place, it may be necessary to reamer the holes $\frac{1}{32}$ in. oversize, and to fit correspondingly larger valves. No attempt should be made to drill the holes large enough to take separate guides, as there would probably not be enough thickness of metal in the main casting to enable sufficiently stout guides to be fitted. If the guides are too thin they would not hold firmly when forced in and would not last long.

The methods adopted for forcing in or removing detachable guides will depend upon constructional features. Screw or small hydraulic presses, if available, can often be utilized. The simplest method, but one requiring special care, utilizes only a special punch, as shown in Fig. 56. Another simple method (Fig. 57) requires a screwed rod or bolt, a washer or bridge piece *A*, and a short length of tube *B*, or a bridge piece *C*.

VALVE SPRINGS

When a valve is put into position with the spring round it, more force is required for compressing the spring than it is convenient to apply by the fingers; valve spring compressors or lifters are therefore necessary. A simple lever, with a forked end resting on a block of wood of suitable thickness as a fulcrum, is often sufficient, but the special tools, of which a number of patterns is available, are generally more convenient for most engines.

Fig. 58 shows a simple U form with a screwed adjustment in connection with a Standard 9 h.p. engine.

Fig. 59 shows the "Millenium," giving a parallel motion of the jaws and operated by a butterfly nut.

Fig. 60 shows the "Nesthill," a simple lever pattern with a hinged strut as fulcrum, alternative length struts being available.

Fig. 61 shows Messrs. Gerrard's "Instanto," which is adjustable for different engines. An aluminium cap rests on the valves, and is connected to the frame by

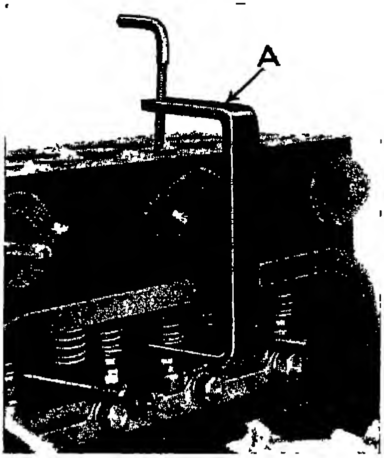


FIG 58 VALVE SPRING COMPRESSOR

linkwork which is self-locking when the spring is compressed. This apparatus holds the valve on its seating, and compresses the spring at one operation, so that the cotter can be at once removed or inserted.

Various forms of cotter or other locking device for holding the spring and valve in position are used

The various parts of the six-cylinder overhead Morris valve are shown in Fig. 62. The small safety clip *A* prevents the valve from dropping through the guide when the spring is removed. Care must be taken when assembling to arrange the retaining caps the

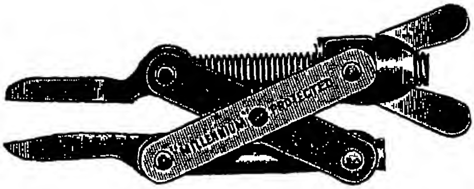


FIG. 59. "MILLENIUM" VALVE SPRING COMPRESSOR

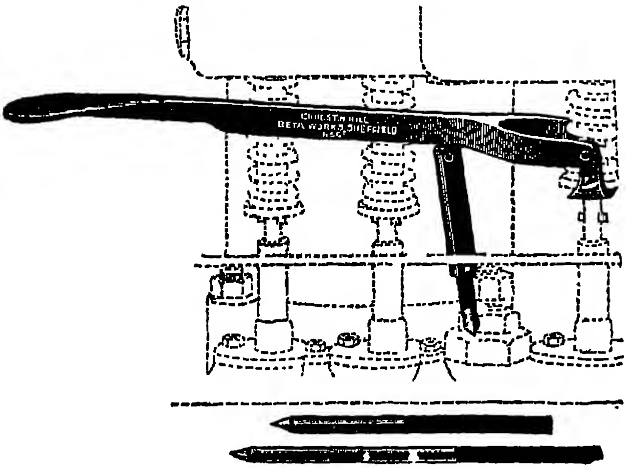


FIG. 60. "NESTHILL" VALVE SPRING COMPRESSOR

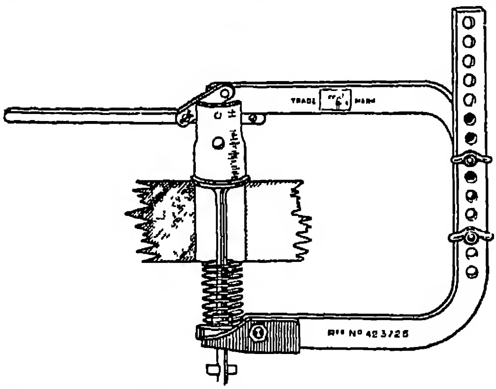


FIG. 61 "INSTANTO" VALVE SPRING COMPRESSOR

right way round, as one of their functions is the retaining of the spring centrally of the valve. The hollow side of the cap *B* receives the end of the spring,

FIG. 62. MORRIS OVERHEAD VALVE PARTS

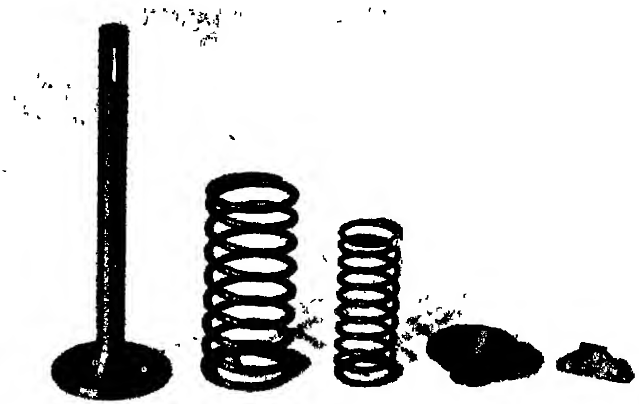


FIG. 63. STANDARD OVERHEAD VALVE PARTS

while the hollow side of the cap *C* abuts against the horse-shoe cotter *D* The hardened steel cap is placed in position last. Extreme care must be taken to ensure that each of the valves after removal is returned to its former position.

In the case of the 14–28 Standard (Fig. 63), the over head valve springs are held by a cotter passing througl a slot in the valve stem In this case, two springs arc fitted to each valve.

The Ford side valve springs (Fig. 64) are held ir place by a combined cotter and end cap bearing on a

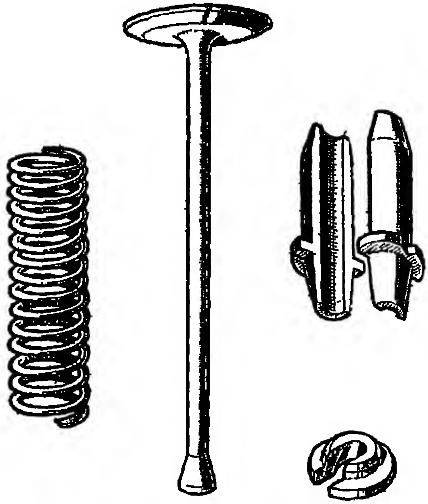


FIG 64. FORD SIDE VALVE PARTS

coned end of the valve stem. This design necessitates a split guide, which is pushed upwards into position after the valve is placed on its seating. Removal and replacement of worn guides is thus very simple.

VALVE TAPPETS

Between the valve-operating cam and the stem of the valve, an intermediate member or valve tappet is always provided The tappet is nearly always provided with adjusting means to ensure that a definite clearance is left between the tappet and the end of the valve stem.

This clearance is commonly, when the engine is cold, ·004 in. for inlet valves, and ·005 in. for exhaust valves, but slight departures from these figures may be made. The makers' recommendations should in all cases be strictly followed. These clearances allow for the increase in length of the valve stems, due to heat expansion when the engine is running. The clearance should be above rather than below those recommended.

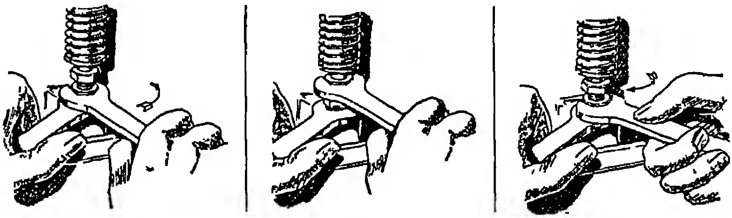


FIG. 65 ADJUSTING MORRIS SIDE VALVE TAPPETS

Left. Loosen the lock nut while holding the tappet by a spanner engaging the flats at its upper end.
Centre. While still holding the tappet, the tappet adjusting screw may be rotated one way or the other to alter the clearance
Right. The lock nut may now be securely tightened, care being taken that the tappet adjusting screw is not moved relatively to the tappet during this operation.

Greater clearance merely means a slight noise, whereas too little clearance may mean that the valve does not bed right down on its seat, in which case the resultant leakage of flaming gases will quickly score the face of the valve and the seat, apart from loss of power.

The Morris valve tappet (Fig. 47) consists of a cylindrical steel body with a disc-shaped end engaging the cam, the tappet being a very nice sliding fit in a hole in the cylinder casting. Into the upper end of the body is screwed the adjusting screw with a hexagon end, and this is held in position, when adjusted, by a lock nut. The method of adjusting is shown and explained in Fig. 65.

In the case of the Star overhead valve (Fig. 48), a

steel ball is interposed between the vertical push rod and the end of the adjusting screw *B*, and care must be taken that this is always in position. To adjust the clearance, slacken the lock nut *A* and adjust screw *B* until the clearance between the end of the rocker and the hard steel valve stem cap *C* is ·003 in., when the engine is warm. The lock nut must be tightened after adjustment.

The clearance should be tested with a steel feeler gauge. In the Star adjustment the ·003 feeler should enter, and the ·004 should not enter.

Some makers fit rollers to the ends of the sliding tappets. Cams of a different contour are then necessary. Tappets in the form of a pivoted lever are sometimes fitted.

Wear between the sliding tappet and its guide will result in noise. Removable tappet guides may be replaced by new guides, the methods of removal and replacement being similar to those employed for valve stem guides. When, as is more usual, the tappets work direct in the cylinder casting, the holes may be reamered $\frac{1}{64}$ in. oversize, and oversize tappets fitted.

VALVE AND IGNITION TIMING

The modern petrol engine works at such very high speeds that the problem of getting the explosive mixture and the products of combustion into and out of the cylinder quickly enough is a serious one. The inertia and lag of the gases is considerable, and, for this reason, the valves in general have to open early and close late, so as to give a longer open period than theoretical considerations would suggest

The design and arrangement of valves, cam gear, ports and passages affect the timing, and hence the recommended timings for different engines varies.

Fig. 66 shows in skeleton form the position of the

crank pin and connecting rod when the several events of the cycle take place. The timing figures given in this diagram are those adopted for the six-cylinder engine of the A.C. (Acedes) car.

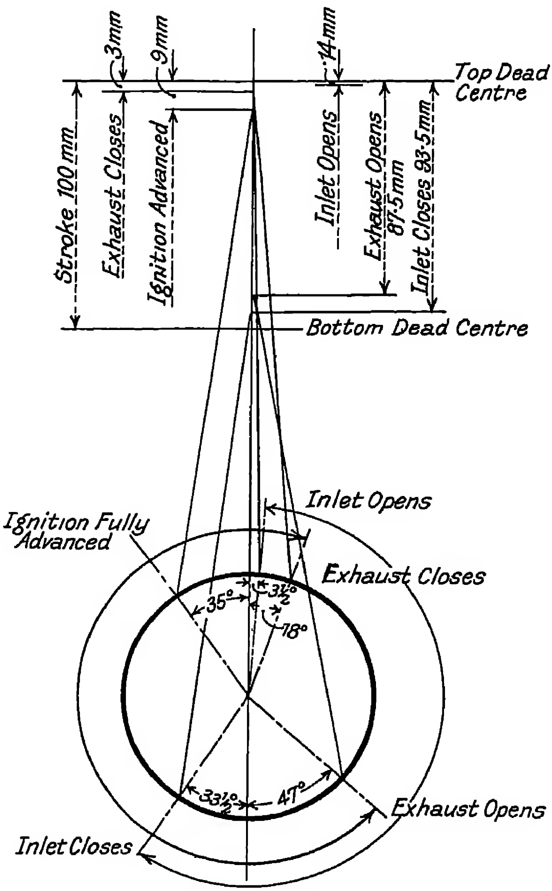


FIG 66 VALVE TIMING ON AC (ACEDES) ENGINE

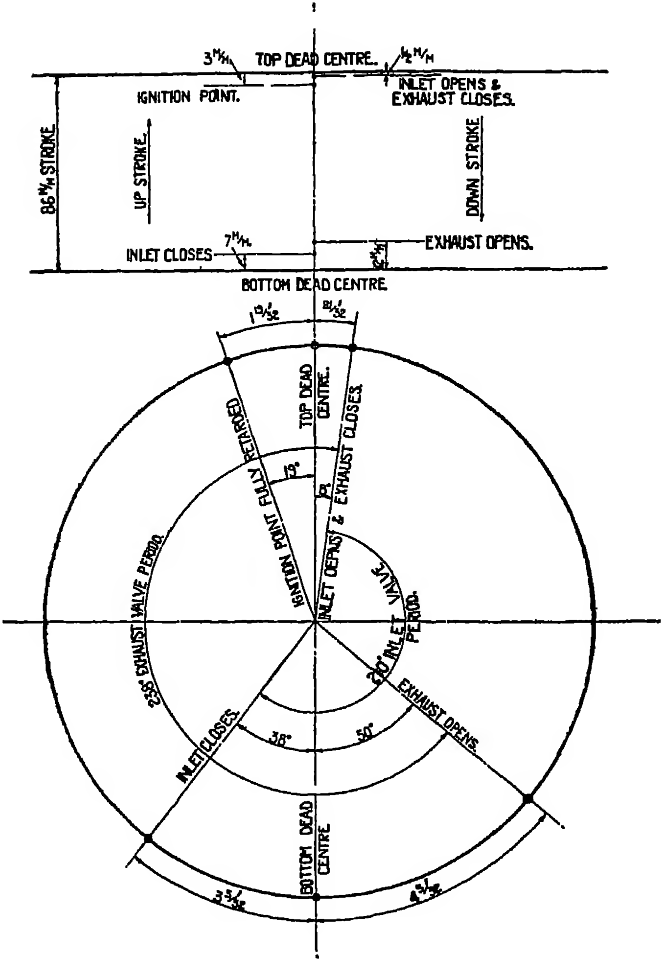


FIG 66A VALVE TIMING DIAGRAM OF SINGER JUNIOR

The inlet valve opens 3½ degrees late, that is, the 'ank has then moved 3½ degrees past the upper dead ›ntre, and the piston has moved ·14 mm. on the down ' suction stroke. The inlet valve remains open during ıe whole of the suction stroke, and during part of ιe compression stroke. It closes 33½ degrees late, the ank pin having turned 33½ degrees beyond the lower ›ad centre. The piston has then moved 6·5 mm. on e up stroke.

Similarly, the exhaust opens 47 degrees early and ›ses 18 degrees late. The inlet and exhaust are thus ›en together, or overlap for a short time.

It will also be seen that the ignition, when fully lvanced, gives a spark 35 degrees early to allow for ›wness of combustion.

The stroke of this engine is 100 mm., and the diagram ves the distance of the piston from its highest posi›n when the different events of the cycle take place.

The makers' instructions must be followed in each se, since the best valve settings for one engine may ›t suit another. A few examples are given below—

Engine	Inlet opens	Inlet closes	Exhaust opens	Exhaust closes
yl. A C (Acedes)	3½° late	33½° late	47° early	18° late
p. Standard .	10° early	50° late	50° early	10° late
rman type 4 MVB	12° late	45° late	45° early	17° late
-50 Star . .	10° early	50° late	50° early	10° late
p Singer Junior	8° early	38° late	50° early	8° late

Methods of driving the camshaft in modern engines fer so greatly that only general principles can be licated.

In some engines with side valves, where the camshaft driven directly at half speed from the crankshaft by

toothed wheels, one of the teeth on one wheel is marked, generally with 0, and the corresponding space on the other wheel is similarly marked. This facilitates reassembly

Where chain gearing is used, whether for a side

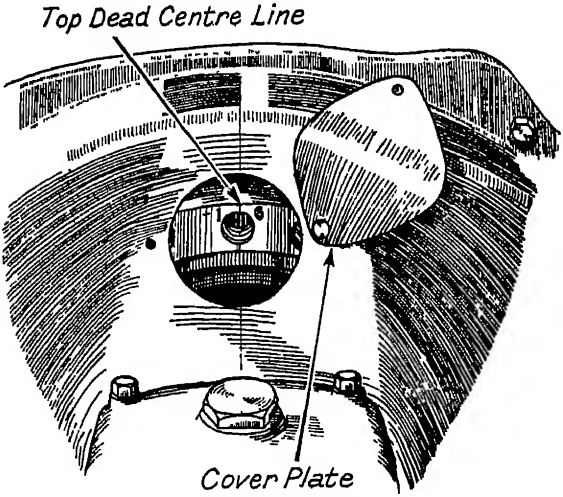


FIG. 67. FLYWHEEL MARKINGS

camshaft or an overhead camshaft, such marks cannot be used.

In most modern engines, the rim of the flywheel is marked to indicate at least the dead centre positions. Fig. 67 shows the system on the 20 h p. Austin, six-cylinder engine. When the marks 1, 6 coincide with the scribed line on the casing, cranks of cylinders 1 and 6 are on their upper dead centres. There are similar marks for the other cylinders. These markings should be made before assembly, when the flywheel is bolted to the crankshaft. It is also desirable that other markings should be made on the flywheel giving its

position, when other events of the cycle occur such as the inlet valve opening.

Consider, for instance, the case of the Dorman engine referred to in the above table. The inlet valve opens 12 degrees late. This corresponds to $1\frac{1}{4}$ in. on a flywheel

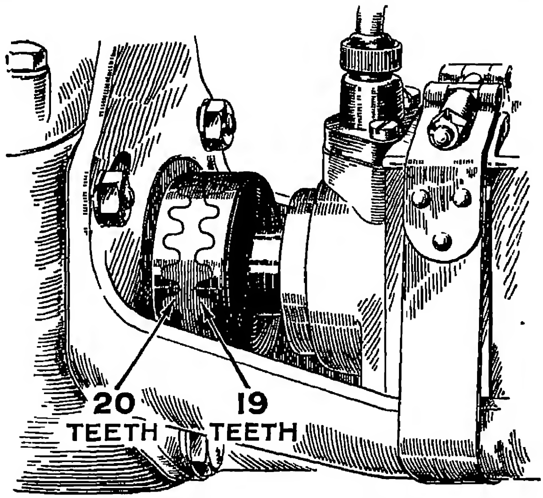


FIG. 68 MAGNETO DRIVE VERNIER COUPLING

12 in. in diameter. A mark $1\frac{1}{4}$ in. behind each of the dead centre marks may therefore be made on the flywheel. If the crankshaft is very slowly turned until the inlet tappet for, say, No. 1 cylinder is just about to lift, the mark referred to should then coincide with the line or pointer on the casting $1\frac{1}{4}$ in. behind the dead centre mark for No. 1 cylinder.

If the correct result is not obtained, the camshaft driving gear must be adjusted as required. This may mean shifting the gear wheels round or moving a chain sprocket one tooth, or making vernier or other adjustments of a gear wheel relatively to its shaft.

Once the correct timing for one event with one valve, such as the inlet opening, is obtained, the opening and closing for all the other valves will be correct, owing to the accuracy of the cam machining.

The tappet clearances must be correct before valve timing is carried out.

The drive for the magneto should be so adjusted that the contact-breaker points are just opening to give a spark in, say, No. 1 cylinder, when No. 1 crank is from 25 to 35 degrees ahead of its dead centre. This should be done with the magneto fully advanced. The Austin vernier coupling magneto drive, shown in Fig. 68, enables fine adjustments to be made.

SECTION XXXIII

GAS ENGINES—FITTING AND ASSEMBLY

BY

MAJOR A. GARRARD, WH EX.

SECTION XXXIII

GAS ENGINES—FITTING AND ASSEMBLY

INTRODUCTORY

GAS engines in general differ from petrol engines in size, and in being of more substantial construction. Speeds are lower and the dimensions of each cylinder are greater, while absolute reliability over long periods of continuous running is essential.

There is much diversity of types and designs. For the majority of gas engines, the four-stroke cycle is adopted and the cylinders lie horizontally. Vertical cylinder engines with various numbers of cylinders working on a common crankshaft and forming a complete unit are, however, often employed.

From the fitting and assembly point of view, much that has been written in the previous section in connection with petrol engines is applicable also to gas engines. Attention will therefore be directed chiefly in this section to features where differences occur.

Single-cylinder gas engines are seldom made in larger sizes than about 150 h.p. Greater powers are obtained by multiplying the number of cylinders, various horizontal, tandem, or vertical arrangements being utilized.

There is little demand at the present day for engines using the ordinary town coal gas in view of the cheaper running costs of heavy oil engines. The demand is, however, still maintained for gas engines using blast furnace gas and suction gas produced from refuse of various kinds. Oil engines for oil producing districts

are also often supplied with a breech end adapted to use natural gas until the supply is exhausted, after which a breech end suitable for oil is fitted to the engine. In the smaller sizes used, for instance, for small electric lighting installations, the engines are made readily adaptable for the use of coal gas, petrol, or paraffin.

As mentioned in connection with petrol engines, modern manufacturing methods aim at the elimination of hand fitting by careful and accurate machining. This aim has been very largely but not fully achieved , it has however, had some influence on repair work also, since it becomes cheaper to use manufacturers replacements made accurately to gauge, than to make and fit new parts

Engines are assembled and run under test by the makers, and are then partly dis-assembled and the parts packed separately for transport ; for instance, pistons, flywheels, connecting rods, crankshafts, or balance weights may be detached and packed separately as required from consideration of size and weight. Many makers stock quantities of parts, so that spares can be readily supplied or engines can be assembled, tested, and dispatched at short notice.

Attention will be directed primarily to what may be described as the ordinary type of gas engine, that is, single cylinder gas engines which, in the larger sizes, require overhead tackle for handling the principal parts.

A general view of such an engine made by Crossley Brothers is shown in Fig. 1, and sectional elevations in Figs. 2 and 2*a*. Details of this engine are referred to in what follows to illustrate the practice described

ENGINE ROOM

The environment of an engine is a matter of the greatest importance, and accordingly a typical model

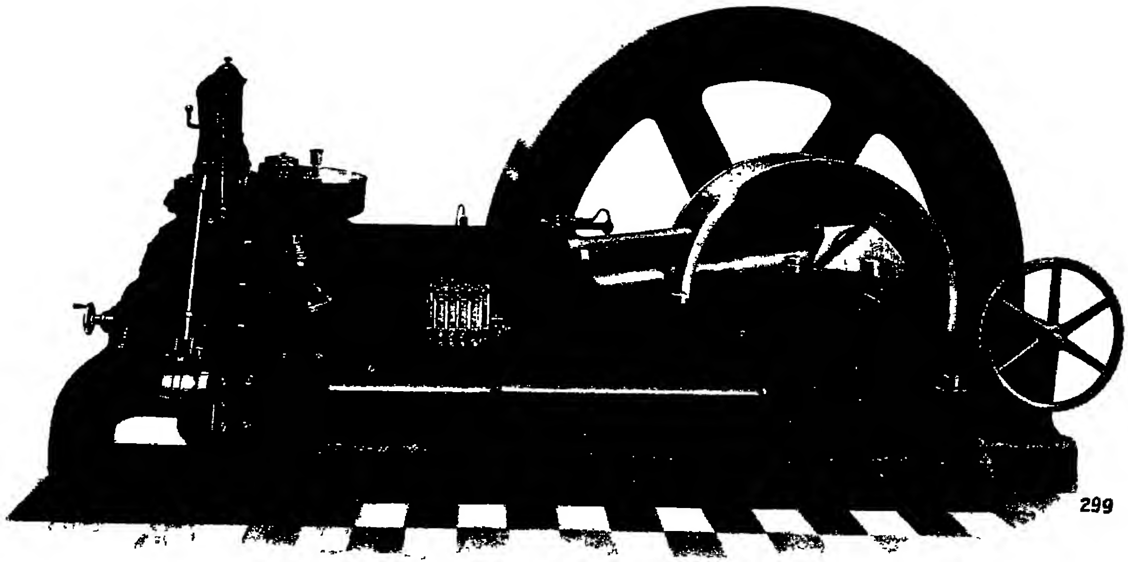


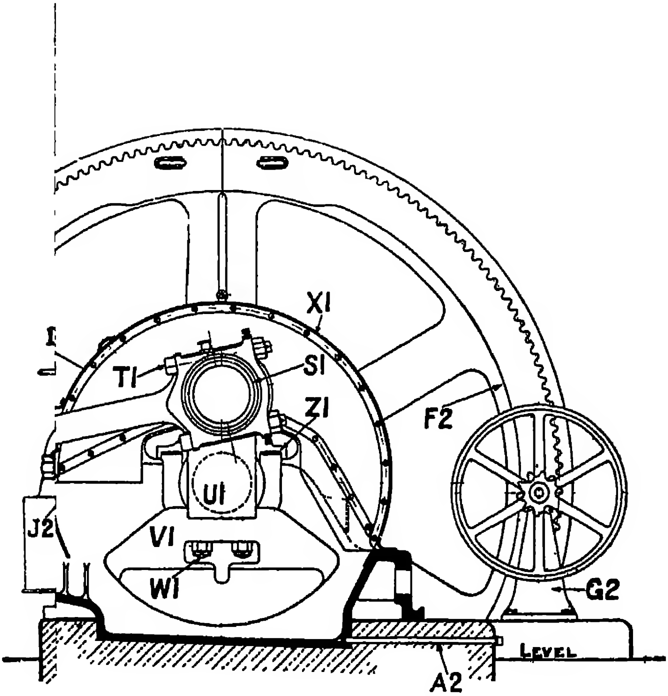
Fig. 1. Crossley Horizontal Gas Engine

engine room is illustrated first in Figs. 3 and 4. The engine must be arranged with ample room all round it, and with its various appurtenances conveniently disposed. It is essential that the engine room should be well ventilated, particularly where suction gas is in use, since it does not smell as strongly as town gas and hence gives less warning, while it is likely to contain more dangerous components or impurities. The various parts and accessories shown will be referred to, as may be necessary, in the following pages.

Vertically over the axis of the crankshaft of the cylinders are arranged two substantial rolled steel joists *B*, each adapted to carry lifting tackle for the erection of the engine in the first instance, and for repair or maintenance work at a later stage. The running gear (not shown) in the case of the transverse or crankshaft girder should run on the lower flanges, so that the upper girder does not prevent a straight run through. The girders and the lifting tackle must be at least strong enough to lift the heaviest engine elements. The gas pipe *A* is connected at some point in its length to a gas bag *C*, having one side of flexible rubber, the object being to make the flow of gas through the meter fairly constant since the suction of the engine is intermittent. Gas engines may be adapted to be worked by petrol, a fuel tank *H* and a carburettor, with a connecting pipe between the two, being then provided.

FOUNDATIONS

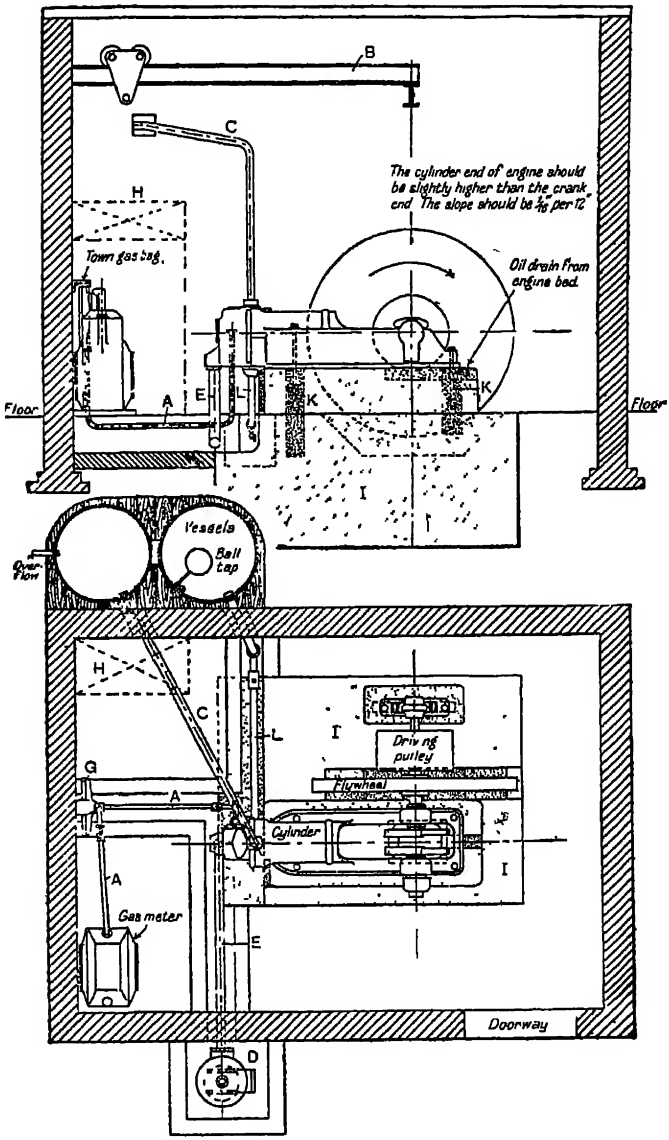
The foundations must be substantial and built on firm solid ground. If possible, it is desirable to go down through made ground to the undisturbed subsoil If the ground is of a yielding character, the foundation should be extended, but preferably a new site should be found. The concrete should be made of 1 part of



s
ve
valve
lubricator
rod
rod, small end bearings
rod, large end bearings
rod bolts

ights
ights studs
ard

*Y*1. Bedplate
*Z*1. Main bearing caps
*A*2. Waste oil drain pipe
*B*2. Water inlet branch
*C*2. Water outlet branch
*D*2. Auxiliary water inlet
*E*2. Auxiliary water outlet
*F*2. Flywheel
*G*2. Barring gear
*H*2. Gas and air block
*I*2. Handle for regulating gas
*J*2. Handle for regulating air



FIGS 3 AND 4 TYPICAL ENGINE ROOM

cement to 3 parts of stones or broken bricks, sufficiently small to pass through a $1\frac{1}{2}$ in. mesh, while 2 to 3 parts of clean, sharp sand to 1 of cement should be added. Brickwork should not be used. The concrete must be set hard before use

The top of the concrete foundation should be left

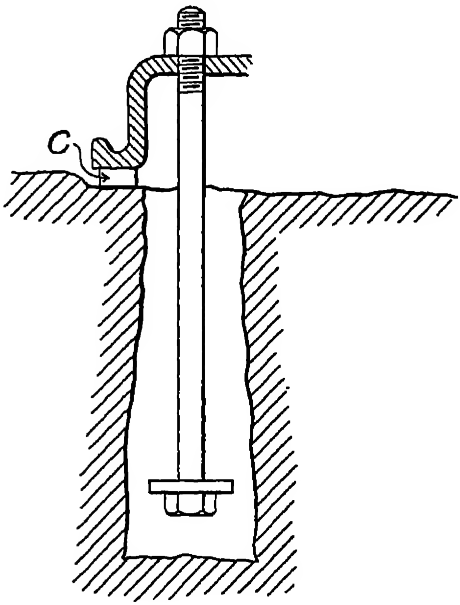


Fig. 5 Holding Down Bolt

rough, and the engine built up on iron packing pieces, about 3 in. × 3 in × $\frac{1}{2}$ in. or larger according to the size of engine. The engine should be levelled carefully, so that it is slightly inclined downwards towards the crankshaft end, to facilitate drainage of used oil from the cylinder, and it should be set correctly with respect to the driven machinery. The foundation bolts should

be suspended in position in the bolt holes by means of their nuts, as shown in Fig. 5, the holes being left tapered and rough so that the cement which is filled in later will grip well.

The iron packing pieces *C* must give a flat wide bearing area, and should be arranged near the bolt holes. They should be sufficient to support the engine independently of the grouting which should be regarded

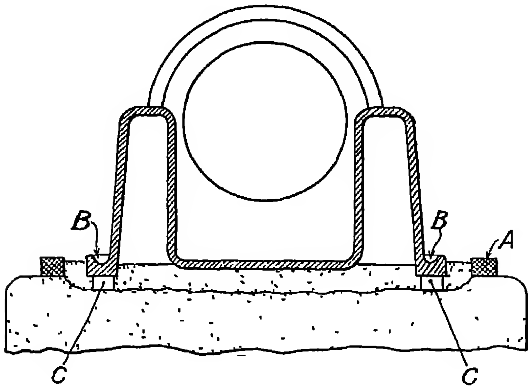


FIG. 6 SECTION OF ENGINE FOUNDATIONS

as a filling. Wedging up by chisels is quite inadequate owing to the very small bearing area. Any extra thickness necessary should be obtained by pieces of sheet metal giving the same supporting area as the blocks.

A clay ridge *A* to retain the grouting, Fig. 6, should be made round the engine bed parallel to the oil groove *B* round the edge of the casting. A board may be used instead of the clay. The bolt holes should then be filled with cement, and cement grouting should be run in level with the oil groove preferably from the inside through holes in the casting, when such are provided for this purpose By this method it is possible to see

whether the grouting has completely filled in the space all round the bed. When the cement has set firmly, the engine bed plate will be uniformly supported all round, and the nuts on the foundation bolts may be tightened up. The oil groove *B* is provided to prevent damage to the concrete, but as a further precaution the concrete may be painted.

When the grouting has partly set, the clay ridge or board may be removed, and the cement rounded or filled up and generally neatly finished. The sole plate under the independent third crankshaft bearing must be similarly mounted on a concrete foundation and secured in position.

The trenches for the exhaust pipes, etc., should be brick lined, and covered with removable iron plates level with the floor. They should be well drained.

CRANKSHAFT ALIGNMENT

In small engines the flywheel, or flywheels, are overhung on one or both sides of the two bearings carried by the bed plate ; but in larger engines a third bearing is provided outside the flywheel and driving pulley. The driving pulley is, however, often arranged outside the third bearing, this arrangement as applied to Crossley engines being shown in Fig. 7. It is essential that all the bearings should be truly in line. Any non-alignment brings about continual bending of the shaft as it rotates. This results in heating and, ultimately, in seizing of the bearings, or in fracture of the shaft. The strongest shaft is not proof against faulty alignment, and will certainly fracture if run under such conditions

The alignment of the three bearings can be tested by measurement of the gaps between the webs of the crankshaft.

The method adopted in the case of the Blackstone

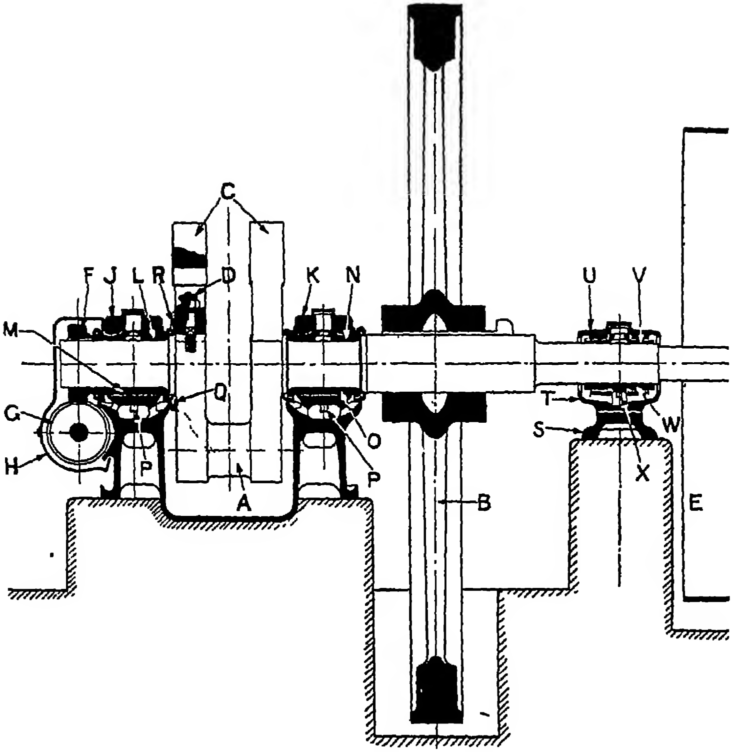


FIG. 7. CROSSLEY CRANKSHAFT, MAIN BEARINGS

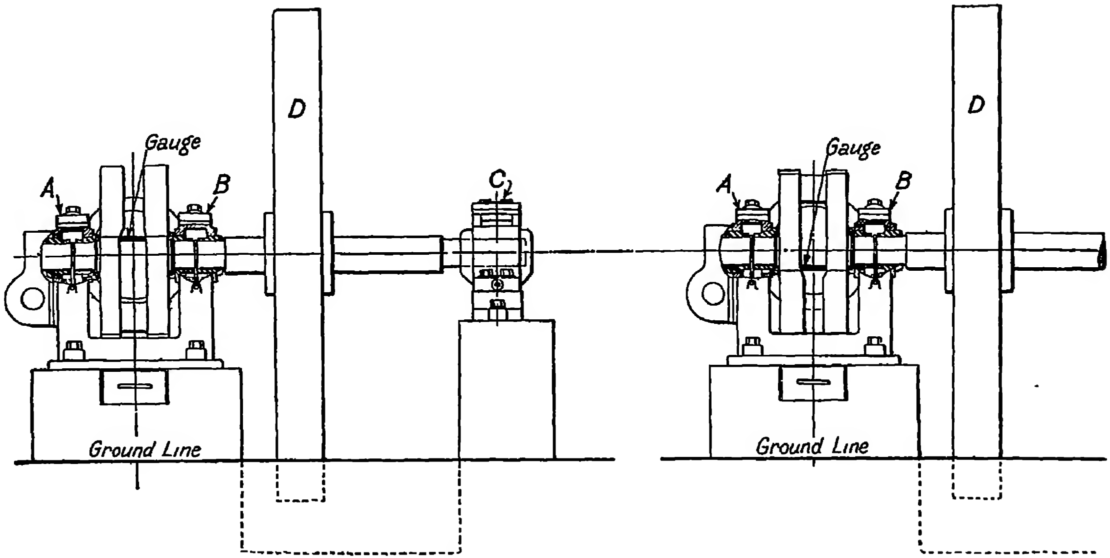
A—Crankshaft
B—Flywheel.
C—Balance weights
D— ,, ,, studs
E—Pulley
F—Crank worm wheel
G—Sideshaft wheel
H—Wheel guard.
J—Main bearing cap sideshaft side.
K— ,, ,, ,, flywheel side.
L— ,, ,, top half, SS side
M— ,, ,, bottom half, SS side
N—Main bearing top half, flywheel side
O— ,, ,, bottom half, flywheel side
P— ,, ,, oiling rings.
Q—Centrifugal lubricator, bottom half
R— ,, ,, top half.
S—Outend bearing sole plate
T— ,, ,, pedestal.
U— ,, ,, cap
V— ,, ,, top half
W— ,, ,, bottom
X—Oiling ring for outend bearing

engines is shown in Figs. 8 and 9, the cranksha
shown supported in the two main bearings A
the engine bed plate, and in the third indepe
mounted bearing *C* disposed outside the flyw
The drawings show the method of testing whe
outer bearing *C* is too high or too low. After
bearings have been bolted down to the foun
the crankshaft (with flywheel) mounted and t
ing caps bolted on, tests are made with a specia
gauge supplied with the engine To test vertic
ment, the engine is turned first into the positio
in Fig. 8, with the crank pin in its lowest posit
the gap between the crank webs is tested v
gauge. The engine is then turned to the
shown in Fig. 9 with the crank pin in its high
tion, and the test repeated If the gauge is
Fig. 8, and tight in Fig. 9, the outer bearing is
too low, while, conversely, if the gauge is
Fig. 8 and slack in Fig. 9, the outer bearing is t

The bearing must be adjusted up or down u
gauge slides easily in both positions A simi
should be made with the crank pin in the tv
zontal positions, to test horizontal alignment,
bearing adjusted if necessary.

These tests should be repeated occasionally a
first erection, since the alignment may be u
settling of the foundations. It is desirable th
should be made at intervals of not more t
months to guard against breakdown.

Messrs. Crossley Brothers advise measureme
gap *X* between the crank webs with an inside
meter, or with a dial micrometer. Fig. 10 sh
a greatly exaggerated manner, the effect due
outer bearing *C* being too high. With the cran
the upper position, the gap *X* is too great, w



FIGS 8 AND 9. METHOD OF TESTING BLACKSTONE ENGINE BEARING ALIGNMENT

gap is too small when the crank pin is in the lower position.

Fig. 11 shows in a similar exaggerated manner the

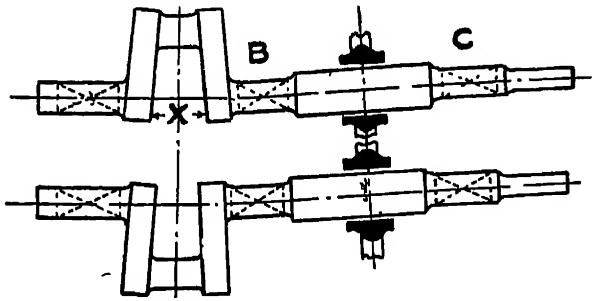


FIG 10. CROSSLEY BEARING ALIGNMENT

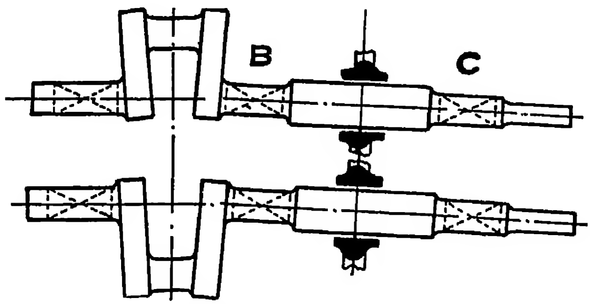


FIG 11. CROSSLEY BEARING ALIGNMENT

distortion of the crank shaft when the outer bearing *C* is too low.

When testing with the inside micrometer, the test points on the webs should be marked. These points should be taken in the middle of the web, and about ¼ in. from the edge.

The dial micrometer should be inserted between the

webs when the crank pin is in the upper position The adjusting nut should then be screwed up sufficiently to prevent it from moving, and the pointer of the gauge adjusted to read at zero. The crankshaft should be given half a turn and the reading of the dial noted. Similarly, readings may be taken when the crank is in its two horizontal positions to test horizontal alignment.

The greatest permissible difference between the measurements, when the crankshaft is in the two diametrically opposite positions, is from three to five thousandths of an inch according to size of engine. If these differences are exceeded, the bearings must be carefully aligned by adjustment of packing between bearing and sole plate, or otherwise according to the means provided.

CYLINDER LINERS

In all except the very smallest engines, the cylinder liner is made independent of the main casting or bed plate. In the Crossley engines, Figs. 2*a*, 12, and 13, the ends of the liner are turned a close parallel fit with the main casting. At the crankshaft end, which is comparatively cool, the joint is made watertight by an india-rubber ring let into a recess in the liner. Soft soap applied to the rubber ring assists when putting the liner into position. At the breech end, where the temperatures are much higher and the conditions much more severe, the joint is metal to metal with, possibly, a very thin smearing of red lead. The liner is held against endwise movement by the breech end, and by the stepped engagement with the main casting.

PISTONS AND CONNECTING RODS

Methods of inserting and removing pistons and connecting rods are shown in Figs. 12 and 13 in connection with the larger Crossley engines

The operations carried out when withdrawing tl piston and connecting rod will first be described, tl reverse sequence being followed when erecting.

The crank is turned to the vertical position with tl balance weights underneath (Fig 12), and a woode bar *A* is rested across the sides of the bed plate 1 support the connecting rod after the big end is di connected This bar *A* should be arranged sufficient far back to allow the balance weights to clear it whe

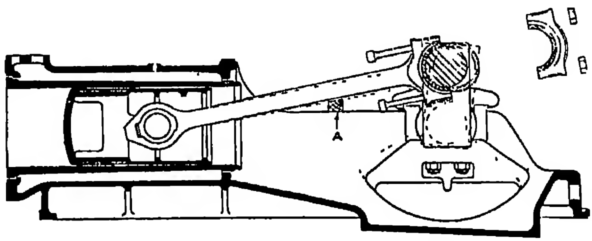


FIG. 12 DISMANTLING CROSSLEY ENGINE

the crankshaft is turned. It should also be clear of tl connecting rod by from $\frac{3}{8}$ in. to $\frac{1}{2}$ in.

The big-end bearing should now be disconnecte The cap and its lining are removed, and the bolts a tapped back as shown, so that they cannot touch a1 damage the journal of the crank pin when the bearii liners are removed

The crankshaft may then be turned slightly and tl back half liner rotated round the journal and wit drawn These operations are shown in dotted lines Fig. 12. Further rotation of the crankshaft results the connecting rod resting on the bar *A*.

It is necessary to lower the crank pin so that tl connecting rod may pass between the crank wel The crankshaft is retained in this position by the woo packing *B* on which the balance weights rest (Fig. 1:

This enables the connecting rod to be lowered into line with the piston, in which position it is secured firmly to the piston by wood packing *C*.

To pull the piston outwards some force may be necessary. This may conveniently be applied by a lever passed through a loop formed in a rope sling *D* passed round the big end of the connecting rod and brought forward between the balance weights. The

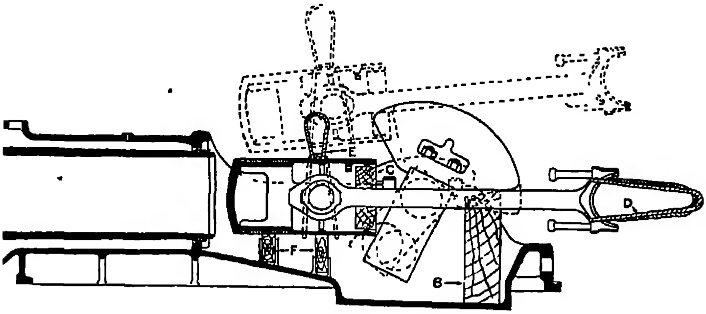


FIG 13 CONNECTING ROD REMOVAL

piston should be withdrawn at first only sufficiently to expose the first piston ring.

Another rope sling *E* is passed round the piston and is secured to the overhead lifting gear, the direction of the pull being slightly forward so as to assist in the withdrawal, the piston being then pulled outwards until further movement is prevented by the crank or weights If, in this position, the weights prevent vertical lifting of the parts, the packing *B* should be removed, and the crankshaft turned until the weights occupy the positions shown in dotted lines in Fig. 13, after which the piston may be lifted clear On the larger engines, ribs are provided on each side of the bed to receive pieces of timber *F* on which the piston may rest when it is withdrawn from the liner

The pistons of gas engines, like all other internal combustion engines, are slightly tapered, being smaller at the back end where the hot gases raise the temperature considerably. Over the forward part, the diameter of the piston is approximately constant, and measures about ·0012 in. less than the liner per inch of diameter. At the back part, the piston tapers down until it is about ·01 in. less per inch of diameter. These are, of course, the differences when the parts are cold.

The piston rings (Fig. 14) are held against rotation

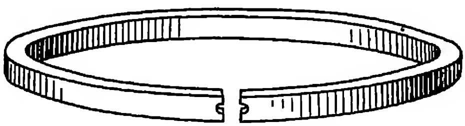


FIG 14. PISTON RING SHOWING PIN SLOT

by pins fixed in the grooves, the pins and the ends of the ring fitting one another without gripping when all the parts are in position. The pins for adjacent rings are spaced about one-third or 120 degrees round the piston.

When removing the rings or putting them in position, they should be sprung over four strips of thin sheet metal arranged longitudinally, as described in connection with petrol engines. Excessive force must not be used as the rings are easily damaged, and will not then fit the cylinder or piston

The big end of the connecting rod is of the marine type as distinct from the locomotive type. Fig. 15 shows a Crossley connecting rod, part of which is in section. The bearings or brasses *D* are of bronze, and generally have white metal facings about ¼ in. thick to give a sufficiently yieldable body. The bearings are bored out carefully, so that their axis is exactly perpendicular to the centre line of the rod, and are then

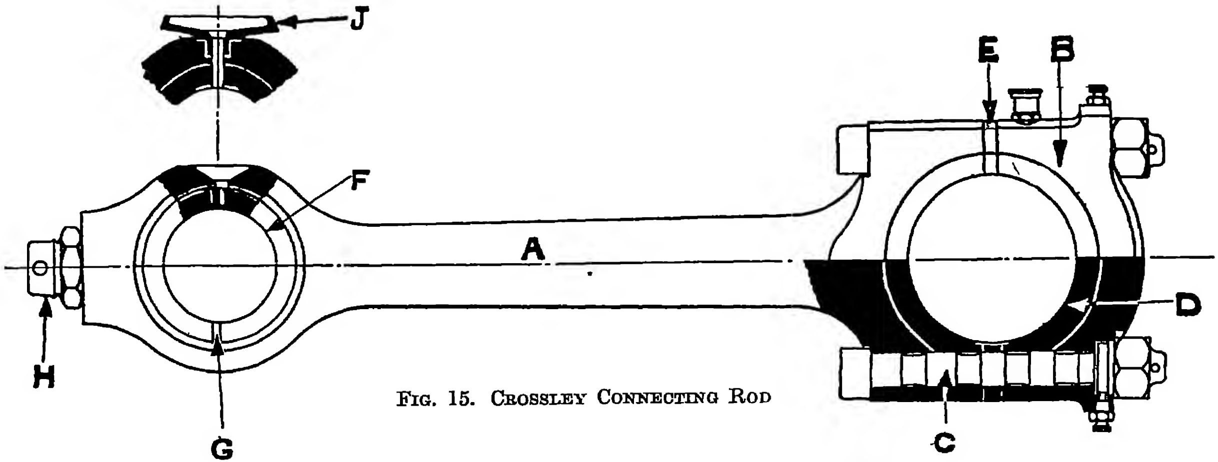


Fig. 15. Crossley Connecting Rod

A—Connecting rod body
B—Large end cap
C—Connecting rod bolts.
D—Large end bearings
E— ,, ,, liner

F—Small end bearings
G— ,, ,, liner.
H— ,, ,, adjusting screw
J— ,, ,, oil trough

fitted to the crank-pin journal by hand scraping, using a colour indicator for the high parts. Proper adjustment of the bearings at all times is essential. They should be quite free to prevent heating, but must not, on the other hand, have enough slack to allow any knocking under the impulses of the working strokes During fitting, the bearings are scraped to make them slacker, and the liners or packing pieces *E* are filed to make the bearings tighter. Alternatively, the ends of the bearings may be filed The bearings must fit the journals properly when the nuts or the bolts *C* are screwed up tightly.

The nuts must not on any account be slacked back to give working clearance to the bearings When tightened up, the nuts are locked by small set-screws, each with its lock nut and by split pins As the clearance is taken up, one or more thin washers must be placed between the faces of the nuts and the split pins.

No engine should be allowed to run with a perceptible knock due to slack bearings, as a great additional load is thereby put on the big-end bolts, breakage of which may lead to a serious and expensive if not dangerous accident. New bolts should be fitted if the bearings have been allowed to knock, or if there has been any seizure of the piston, or in any case after 10,000 hours running or, say, three or four years use. The bolts must be tightened up equally so as to distribute the loads properly, and care must be taken to ensure that they are not twisted or lengthened. If kept bright, a crack can be detected at an early stage. To facilitate examination, Crossley big-end bolts (Fig. 16) are marked along their length with a line, and with a centre punch mark *B*. A straight edge may be placed along the line to detect twist, or the bolt may be mounted in V blocks and a scribing block used for the same purpose. The makers supply with their engines

articulars of the maximum permissible elongation and wist. For example, on a bolt having the dimension l equal to 9 in., the maximum extension permissible s ·018 in., and the maximum twist is $\frac{1}{16}$ in. When the olt has been deformed to such an extent that either f these figures has been exceeded, new bolts must be tted

Connecting-rod bolts, if of nickel or other special teel, should not be annealed Such bolts are made from pecially heat-treated steel to obtain the best combina-ion of strength and toughness. When replacements are

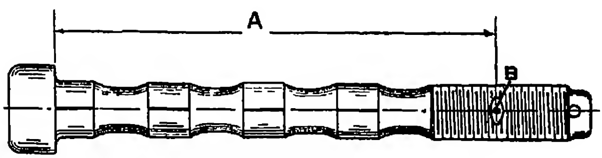


FIG. 16 CROSSLEY BIG-END BOLT

necessary, new bolts should in general be obtained from he makers If, however, the bolts are made of York-shire iron or mild steel, they should be annealed about once a year to reduce risk of crystallization and consequent breakage.

The small end bearings are of brass, and are sometimes livided like the big-end bearings, but in the case of he Crossley engines, Fig. 15, the liners are placed in a ring-shaped end to the connecting rod. As the amount of movement in these bearings is small, and he chances of heating correspondingly less than in the case of big-end bearings, white metal is rarely employed. The gudgeon pin is held against rotation in the piston and the bearings are fitted to it, as in the case of the big end, with freedom but without slack. The bearing surfaces are scraped to slacken, and the abutting ends

of the bearings or brasses are filed to tighten the bearings. Alternatively, the liner *C* may be filed. The bearings are tightened up firmly by the adjusting screw *H* provided with a lock nut

BEARINGS

The crankshaft or other bearings are fitted with liners or brasses of bronze, or of bronze lined with white metal. These bearings are all fitted and finished substantially the same as the big-end bearings referred to in the preceding section. It is therefore unnecessary to do more here than to emphasize the necessity of careful fitting to obtain a large bearing area and accurate adjustment to avoid either knock or tightness. Lubrication will be dealt with in the lubrication section.

FLYWHEELS

The flywheels of gas engines, and particularly those with one cylinder only, are very heavy to ensure uniform speed of rotation. The flywheel must run true and be absolutely rigid with the crankshaft. Further, since flywheels are always made of cast iron, care must be taken in fitting to the crankshaft to avoid strain or distortion.

Flywheels are of three types—

1 Solid, generally with curved arms to avoid strains, due to unequal cooling of the different parts of the casting.

2. Solid rim and split boss.

3. Split rim and boss

In the solid type the boss is bored accurately to gauge, and the portion of the crankshaft on which the flywheel fits is accurately machined to obtain a tight fit between the two.

The flywheel is drawn on to or off the shaft by a

screw pulling gear, the arrangement recommended by Crossley Brothers being shown in Fig. 17. The flywheel is first placed upright on the floor or preferably is held by overhead lifting gear, and the shaft is entered as far as possible. To ensure correct relative position, the

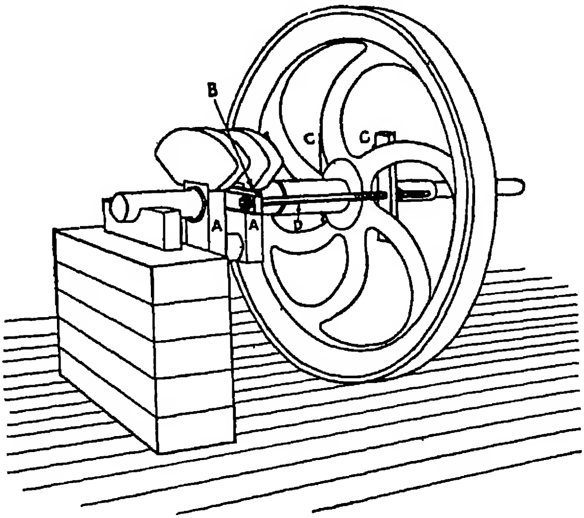


FIG 17. FLYWHEEL PULLING GEAR

key or keys should be entered at the first opportunity. The end of the crankshaft should be supported, as for instance, by a Vee block and packing case, bench or other support. The pulling gear is then rigged up as follows: A plate is arranged across the near crank web *A*, and is connected by two long bolts *D* to two plates *C*, each arranged across two arms of the flywheel. The bolts must be screwed for a considerable part of their length. The nuts must be tightened up uniformly, putting an equal tension on the two bolts as far as

possible until the flywheel is drawn on to its correc
position. The keys should then be driven in tightly.

When removing the flywheel the operation is simila
but the bar is placed across the end of the crankshaf
remote from the crank, and the plates are arrange
across the flywheel arms on the side near the crank.

In the case of flywheels having a divided boss, th

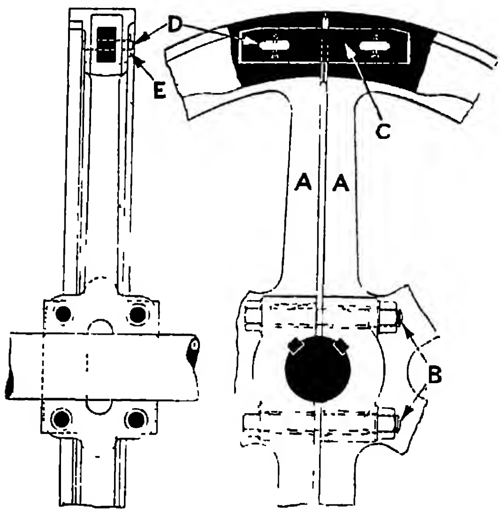


FIG. 18. FLYWHEEL WITH SPLIT RIM AND BOSS

boss is bored out to fit the shaft accurately, so that it
may be clamped tightly on to the shaft without strain-
ing the casting. The flywheel and shaft may be held
as described previously, but it is necessary first to
slacken the bolts holding the parts of the boss together
and to separate slightly the two parts of the boss by
driving iron wedges between them. This enables the
shaft to be entered easily without any pulling gear
The keys may then be entered to position the flywheel,

the boss bolts tightened up, and the keys driven in tightly.

Fig 18 shows a large Crossley flywheel in two separate halves, as fitted to the larger single-cylinder engines and to all double-cylinder engines.

The meeting surfaces of the rim must be scrubbed clean with paraffin, to ensure close metal to metal contact before final assembly.

One of the two halves is lifted into the flywheel pit and is packed up to its correct level, and carefully arranged in position to suit the crankshaft which is then lifted up and lowered into its bearings. The dowels *C* are then placed in position in the holes in the lower half. The upper half of the flywheel is then lifted and lowered carefully into position, and the wedge-shaped cotters *D* inserted and tapped lightly in. The keys are next inserted into the keyway an inch or two, that is, only sufficiently to ensure that the flywheel is in its correct angular position The boss is next clamped firmly on to the shaft by the boss bolts *B*, and the cotters may then be driven home finally and secured against slacking back by split pins *E* The flywheel keys are then driven right in.

KEYS AND KEYWAYS

All keys and flywheel keys in particular must be driven up tight, and must be kept tight A slack key will sooner or later certainly cause breakage of the flywheel or the crankshaft or both. It is impossible to pack up a slack key with liners. A new key must be fitted. Keyways are generally machined parallel, and care must be taken that they are not burred or otherwise rendered smaller at the entering end. This defect is shown in Fig 19 in an exaggerated manner. The key *A* is slack for its whole length, except at the entering

point. The keyway may be made slightly taper to ensure tightness when the key is driven in.

The key must fit at the sides and at the top and bottom, as shown in Fig. 20. No slackness whatever is permissible, except at the parts indicated in dotted lines. A key may be filed slightly hollow at this part when a keyway has been damaged, for instance, by a slack key, but it is better to true up the keyway. If there is any slackness whatever, so that a key starts rocking or working, it will ultimately burst the boss or do other serious damage Fig. 21 shows a key which bears only at the middle of the top and bottom keyways, and will ultimately cause a breakdown The figure shows this defect exaggerated, but it must be clearly understood that the smallest clearance or slackness towards the corners will cause trouble. Fig. 22 shows a loose key taking a one-way drive.

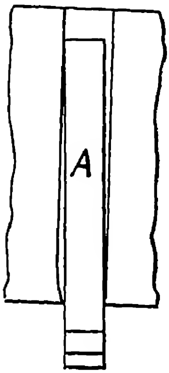


FIG 19. SLACK KEY

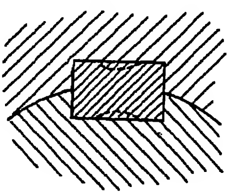


FIG. 20. PROPERLY FITTING KEY

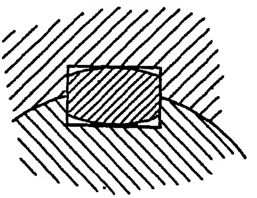


FIG 21. BADLY FITTED ROUNDED KEY

With a new engine, keyways and keys are generally machined carefully to size, so that there is very little fitting, but in carrying out repair work hand fitting may be necessary. To fit keys properly, practice

essential, but some guidance may be furnished
.reful attention must first be given, as suggested
ove, to the keyways. The key must be of hard steel
its corners are subjected to severe compressive forces,
d they are not well backed up by a mass of metal.
ugh axle or similar steel is satisfactory. For small
ys, an old file which has been annealed is quite suit-
le. The key is fitted by a succession of trial entries.
r each trial it is driven in lightly and then removed ;

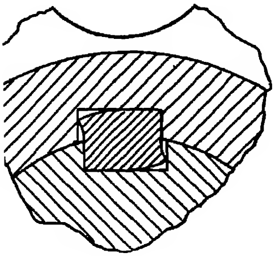


FIG. 22. SLACK KEY

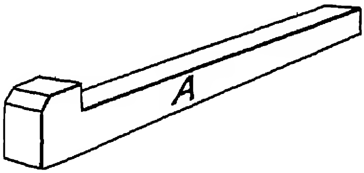


FIG. 23. SHAFT KEY

ead as shown in Fig. 23 being useful for this purpose.
terial is then filed off in accordance with the mark-
s to ensure proper engagement along the whole
gth. One or two inches, according to the size of
key, should be left for the final drive home The
and keyways should be covered with thick gear
which will help to prevent seizing of the metal and
l facilitate removal.

BREECH ENDS

'he breech end carries the inlet and exhaust valves
the ignition block, while the interior forms the
ibustion chamber. Fig. 24 shows a sectional trans-
se view, and Fig 25 a side view of the Crossley
ech end. The joint between the front face of the

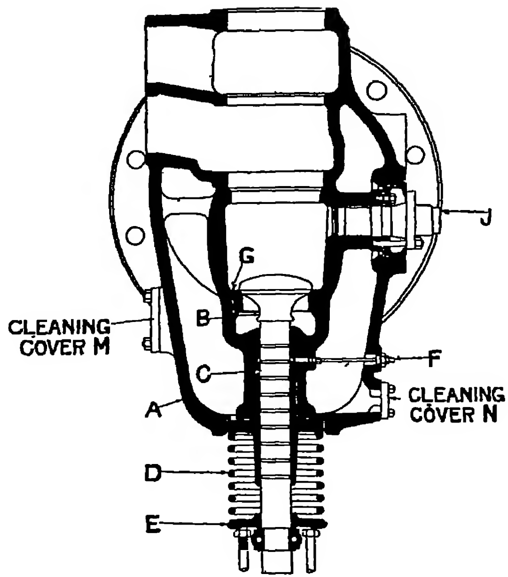


Fig. 24

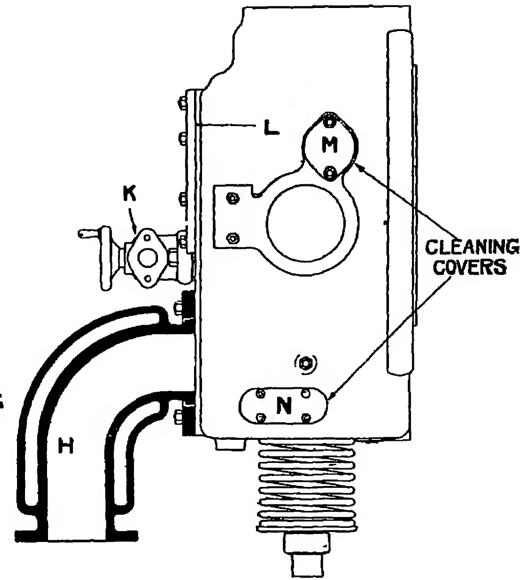


Fig 25

Crossley Breech End

A—Breech end.
B—Exhaust valve.
C— ,, ,, guide.
D— ,, ,, spring
E— ,, ,, ,, plate.

F—Exhaust valve oil pipe
G— ,, ,, seating (not used)
H—Exhaust bend (water cooled)
J—Igniter block

K—Starter valve.
L—Breech end cover
M—Cleaning cover.
N— ,,

reech end and the ends of the main castıng and ylinder liner is made with thın sheet asbestos. The ampbell Gas Engıne Co. recommend for their engines est sheet asbestos soaked ın boiled linseed oil and neared with blacklead.

The nuts on the studs securing the breech end in osıtion must be screwed up carefully to avoid distor- on and unevenness of the joint. All the nuts should ə gone over a number of times, tıghtenıng each up ıly a little at a time, say, one-quarter to one-half a ırn, so that all are gradually tightened up uniformly ı is advisable also to tighten up opposıte nuts in ıccession ; thus, first a top nut, then a bottom nut, əxt a side nut, and then an opposite side nut, and so ı.

These remarks on the bolting up of joints apply also ı the several other smaller joints on the breech end, hich have to be made from time to tıme. The copper- bestos jointings frequently used will be unevenly ımpressed if tightened hard at one side before the her is held, and will not seal the joınt properly.

Fıg. 26 shows a sectional view of the Hornsby- ockport breech end with valves.

VALVES

The exhaust valve is generally arranged at the lower le of the breech end, and opens upwards while the let valve is vertically above and opens downwards.

The Crossley exhaust valve ıs shown in section in Fig. , and also in Figs. 2, *2a* and 27. In this construc- ɒn, the exhaust valve, which is generally of cast ıron, əts directly in the breech end or in a detachable ət *G*, and ıs guided by a detachable guide *C*. The ide is bolted up in positıon, and the valve lowered ɔo ıt from above through the openıng ın whıch the et valve cage ıs located when in position. The valve

is held by a T-ended grinding spindle which is screwed into a hole in its upper surface. To compress the valve spring, the attachment shown in Fig. 27 may be used. The spring *O* and plate are placed in position, and the

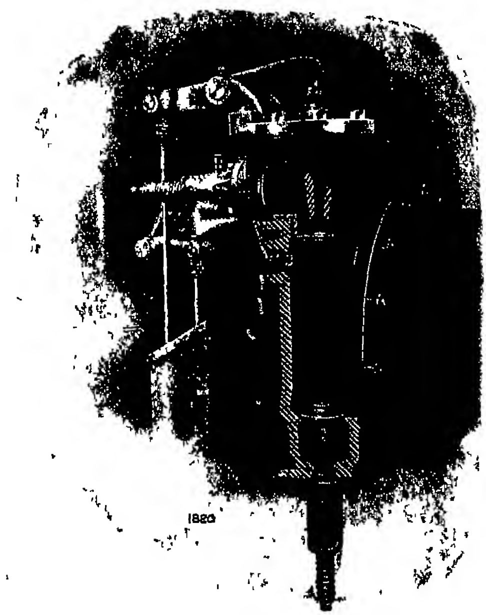


FIG 26. HORNSBY-STOCKPORT BREECH END

spring compressed by nuts on studs *N* which are screwed temporarily into holes on the underside of the breech end or the flange of the guide. When the spring is sufficiently compressed, the nuts *R* are screwed on to the valve stem, after which the studs *N* may be removed. The nuts *R* must be screwed up until the pull of the spring on the valve prevents lifting when

ınning without load, except, of course, when operated 7 the cam gearing.

The Crossley inlet valve *B*, Fig. 28, is carried with 3 sprıng *D* ın a detachable block *A*. The gas valve is slidable on the main valve shown. It is pressed ›wards by a spring *F* against nuts which must be so

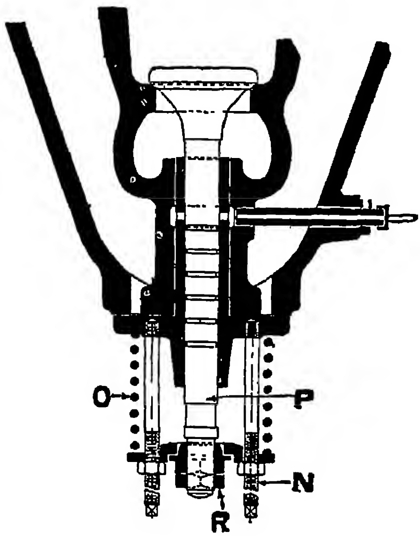


Fig 27 Crossley Exhaust Valve

justed that the gas valve opens slightly after and ›ses slıghtly before the valve *B* to prevent leakage gas into the aır suction silencer. Both valves are awn upwards on to their seats by a spring *D* acting a guıde *R*. The valve is opened downwards agaınst ɜ sprıng *D* by a cam *C*, rod *M*, and rocking lever *G*. e fulcrum or pıvotıng point of the lever *G* consists the end of the approximately vertical arm of the lt-crank lever *H*. The governor, through a link *K*,

controls the position of the fulcrum, and hence varies the lift of both the gas valve *E* and the mixture inlet valve *B*.

The valve spring *D* is compressed when assembling,

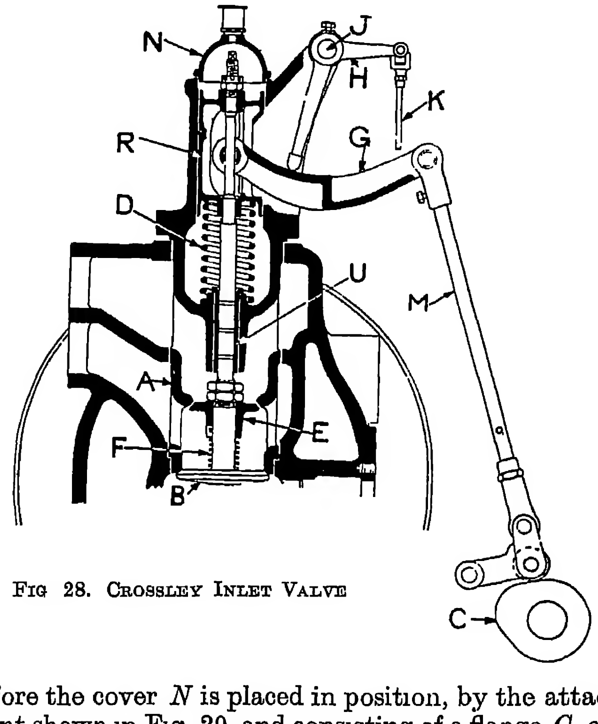


FIG 28. CROSSLEY INLET VALVE

before the cover *N* is placed in position, by the attach ment shown in Fig. 29, and consisting of a flange *C*, and two studs adapted to be screwed temporarily into the head of the admission valve block The flange is screwed down, forcing the valve guide *H* down, and compressing the spring until the nuts *J* can be screwed on to the valve stem.

The valves are ground in by oiling the faces and

sprinkling emery powder thereon. Carborundum should not be used for this purpose. The T-headed grinding spindle referred to above is then screwed into the valve and locked by a nut. The valve is pressed on to its seat lightly and twisted to and fro, being lifted at intervals, and turned to a different position. Next

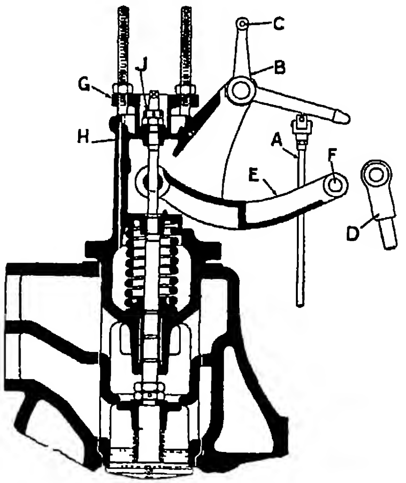


FIG 29. COMPRESSING INLET VALVE SPRING

remove the valve and wipe it and its seating quite clean Then replace it, and work it round lightly on its seat while dry The valve and seating should, if properly ground in, show a continuous engaging surface all round. If this result has not been obtained, the grinding should be repeated as many times as may be necessary.

VALVE OPERATION

The usual side shaft shown in the Crossley general arrangement, Fig 1, and also in Fig. 30, is mounted

in two bearings, one at each end. The shaft is arranged below the level of the crankshaft, and is driven by skew or worm gearing, the driven wheel *J* being keyed on the overhung front end of the side shaft. The lower gear wheel dips into an oil bath, while the adjacent bearing is lubricated by the usual ring oiler, in which a ring substantially larger than the shaft runs thereon and dips into an oil bath.

A somewhat similar example of this class of driving

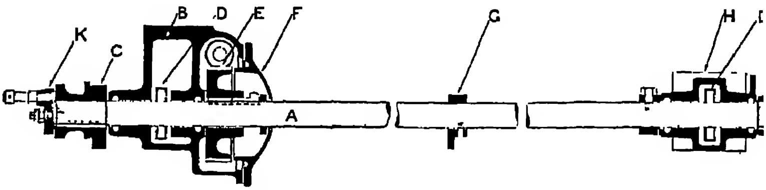


FIG. 30. SIDE OR CAM SHAFT

A—Side shaft
B— „ bracket, cylinder end.
C—Combined cam.
D—Oiling ring.
E—Governor wheel.
F—Cover for side shaft bracket.
G—Lubricator cam.
H—Side shaft bracket crank end.
J— „ wheel.
K—Driving disc for electric ignition.

mechanism as applied to a Tangye gas engine is shown in Fig. 31.

Referring again to the Crossley side shaft, Fig. 30, a gear wheel *E* keyed to the shaft drives the governor spindle, and a combined cam *C* operates both the inlet and exhaust valves.

When erecting, care must be taken that the worm gearing for the side shaft is properly engaged. A tooth on one of the wheels is invariably marked to engage the gap, which is also marked between two teeth on the other wheel. In general, each wheel is marked *O*, and the marks on the two wheels must coincide when the two are rotated to a suitable position. The markings used on Crossley engines are shown in Fig. 32.

Owing to the speed at which gas engines run, and the inertia or lag of the incoming mixture and the exhaust gases, the valves are set to open considerably before and close after the dead centre positions. The engines of different makers differ somewhat as regards timing, owing to variations in design

It may be necessary on new or on repair work to

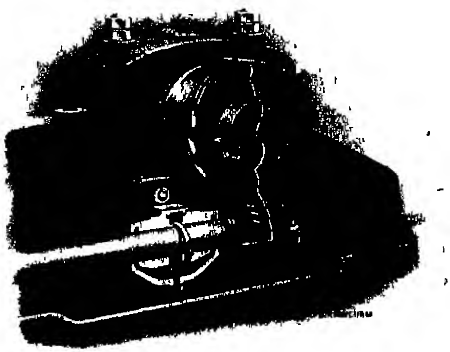


FIG 31. TANGYE CAMSHAFT DRIVING GEAR

check or to reset the valve timing, and the following will explain how this is done.

Fig. 33 shows diagrammatically the positions of the crank pin when the different events of the cycle take place, the timing shown being that adopted on Crossley engines

Consider as the starting point of the cycle, the position of the flywheel when the crank pin is at A on its dead centre at the beginning of the suction stroke. With the ignition switched off, and the gas and starting valves closed, turn the flywheel through 30 degrees to B, when the exhaust valve should just have closed

and the exhaust roller should only just be leaving the exhaust cam. Then turn the flywheel to *C*, where the inlet valve should similarly just be closing.

Next turn the flywheel to *D*, when the magneto trip lever should be released. Further rotation should

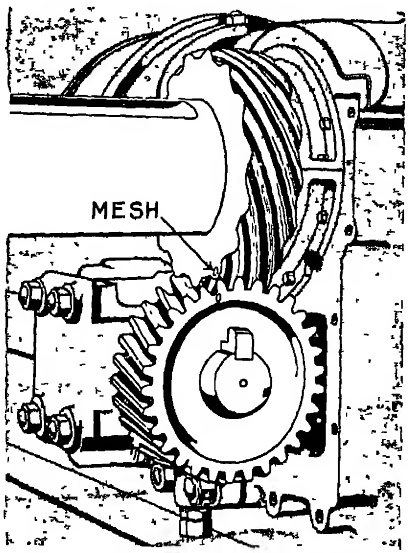


FIG 32. CAMSHAFT DRIVING GEAR

show the exhaust valve opening at a point on the working stroke, when the crank is 55 to 60 degrees ahead of the outer dead centre. The admission valve next opens when the crank is at *E*, 30 degrees ahead of the inner dead centre. From *E* to *B* there is an overlap period, when both air inlet and exhaust valves are open. It should, however, be remembered that the gas valve opens after the main admission valve, so that only air enters at first.

The admission valve will only open and close at the

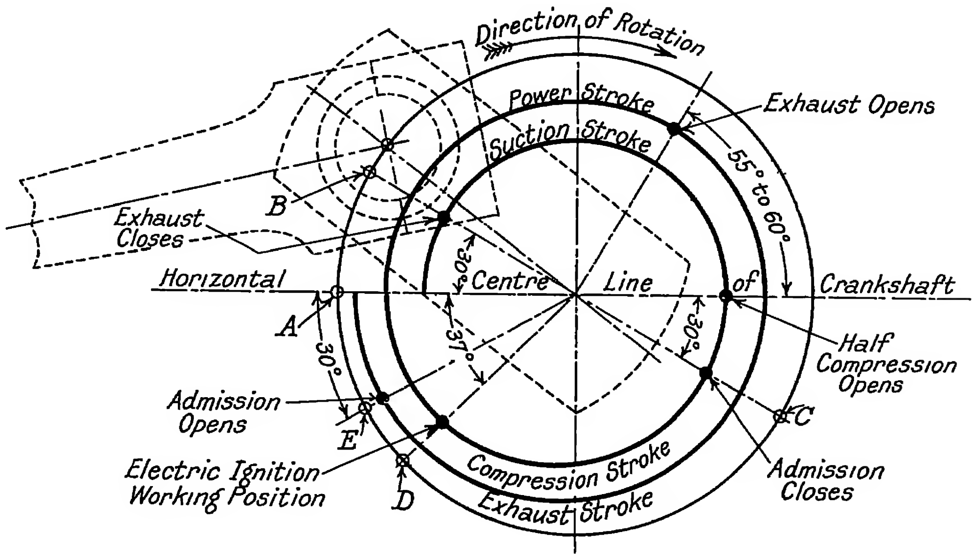


FIG 33 CROSSLEY TIMING DIAGRAM

correct times if the clearance between the radius lever *G*, Fig. 28, and the end of the upright arm of the fulcrum lever *H* is maintained between $\frac{1}{64}$ in. and $\frac{1}{32}$ in. for all positions along the radius lever. The upper limit given should not be exceeded and, on the other hand, the fulcrum lever must never be allowed to stick, as the governor would then be out of action, and the speed might increase to a dangerous extent.

The clearance between the exhaust cam and the exhaust roller should be kept at $\frac{1}{32}$ in. to ensure correct timing of the exhaust valve.

LUBRICATION

Modern machinery depends for its successful operation upon lubrication. Metal surfaces cannot work together and carry the necessary heavy loads, unless they are separated by a thin film of oil having the necessary body to prevent it from being squeezed out under the conditions of load, speed, and temperature to which the bearing or other part may be subjected.

The cylinder walls and the stem of the exhaust valve work under the most severe conditions, and provision generally is made to supply them with oil continually from a reservoir by means of pumps. In larger engines, the big-end bearing and the small-end bearing or gudgeon pin are similarly supplied with oil. A number of small plunger pumps, one for each of the parts in question, are worked from the cam shaft and deliver oil through sight feed glasses, the flow to each part being individually adjustable.

The Crossley central mechanical lubricator comprises four pumps supplying oil respectively to the cylinder, exhaust valve, big-end bearing, and gudgeon pin, one of the pumps being shown in section in Fig. 34.

The filling opening to the container *A* contains a wire gauze filter, which must be examined for soundness

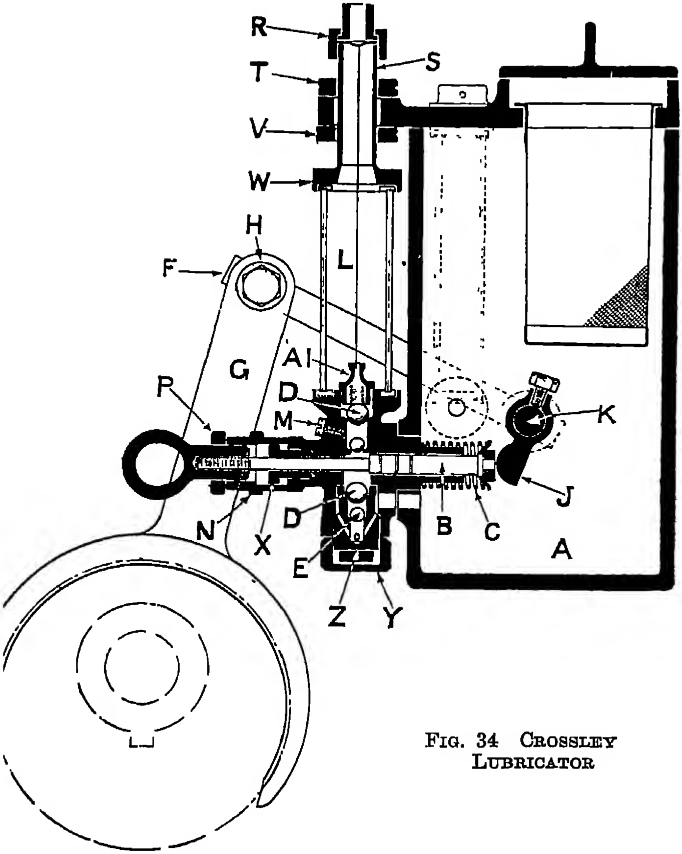


Fig. 34 Crossley Lubricator

A—Container.
B—Plunger.
C— ,, spring
D—Ball valve, large
E— ,, ,, small.
F—Driving arm.
G—Operating rod.
H—Connecting rod pin.
J—Internal tappet lever
K— ,, ,, ,, layshaft.
L—Sight feed glasses containing salt water.

M—Vent screw
N—Adjusting sleeve
P—Lock nut for adjusting sleeve
R—Union nuts for oil pipes.
S—Shank connections to feed glasses.
T—Lock nut for shank *S*.
V— ,, ,, ,, *S*.
W—Joint washers for glasses
X—Gland ring for plunger
Y—Cap for suction valve cage.
Z—Cage for suction valves.
*A*1—Delivery nozzle

when assembling. The spindle K enters the container through a bearing which must be carefully packed, and is oscillated by an arm F and an operating rod G, worked from a cam or eccentric at about the middle of the length of the camshaft. To the shaft K are secured levers J, one for each pump. Each plunger B must fit

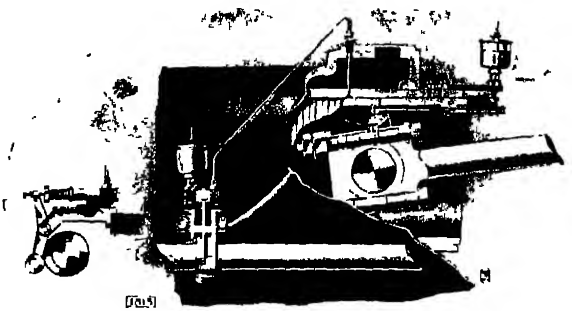


FIG. 35. HORNSBY-STOCKPORT LUBRICATION OF CYLINDER, PISTON, AND GUDGEON PIN

its cylinder without any perceptible slack. An extension of the plunger is fitted with a finger ring, by means of which each pump may be worked independently The extension also carries a sleeve N, which acts as a stop and so limits the effective stroke. The gland X for the extension requires special attention, as it has to stand the full delivery pressure of the oil.

The suction and delivery ball valves may be tapped lightly through a brass rod on to their seats, which are of much softer material so as to make them fit properly. The sight-feed glasses L should contain a saturated solution of salt, and must be clamped firmly endwise, care being taken that the leather washers at the ends are sound

The small copper pipes which lead the oil to the veral points where it is required must be filled with before starting. Before the joints at the ends remote om the pumps are tightened up, the pumps should be orked by hand until oil appears at said ends. All e pumps may be worked together by the lever *F*,

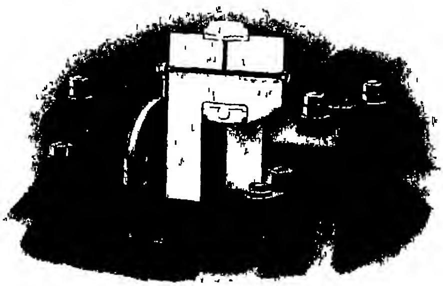


FIG. 36. TANGYE CENTRIFUGAL OILER FOR CRANK PIN

one only at a time by its finger ring without the ving gear being disconnected.

Fig. 35 shows, in the case of a small Hornsby-Stock-t engine, how the oil is supplied to the cylinder walls l piston, and also to the gudgeon pin bearing. In latter, oil from a drip-feed reservoir or from a np is supplied steadily to a wiper, from which it is iveyed by a conduit inside the piston to a point ove the gudgeon bearing and drops thereon as shown. e parts of the wiper must touch only lightly or, ferably, not at all, provided that they are sufficiently near to transfer the oil drop by drop.

The oil connection *F* to the stem of an exhaust ve is shown in Fig. 24.

Fig. 36 shows a centrifugal oiler for the crank] of a Tangye engine. The oil supplied drips into a ri which turns with the crankshaft, and is concave hollow on the inside. The oil, under the action centrifugal force, flows outwards to the crank-] bearing through the pipe and drilled oilways shown

For the various other parts of an engine, ring oile

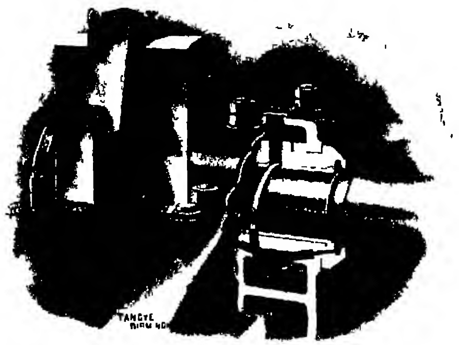


FIG 37. TANGYE RING OILERS FOR CRANKSHAFT MAIN BEARINGS

drip lubricators, or merely oil holes or nipples ar provided.

Figs. 37 and 31 show ring oilers applied to Tangy engines in connection, respectively, with a crankshaf bearing and a camshaft bearing adjacent to the crank shaft. The rings must be large enough to dip wel down into the oil reservoir, and must be quite free anc loose. Provision for the return of surplus oil to the reservoir will be noted.

The several copper pipes must be laid neatly in position and clipped as required, sharp bends being avoided.

Considerable care is necessary in the final assembly.

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